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# POWER FLATTENING TECHNIQUES FOR RADIOISOTOPIC THERMOELECTRIC GENERATORS

Final Report
July, 1984



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## SUMMARY

The objective of this program was the investigation of a novel means of reducing the potential ecologic hazards that may be asociated with radiosotopic thermoelectric generators (RTG's). A number of short lived isotopes have lower toxicities and are more ecologically acceptable than the Plutonium 238 used at present. In addition, the shorter half lives significantly reduce the time period during which isotope encapsulation must be assured ($\approx$10 half lives).

The technical approach involves the use of a gas controlled heat pipe to maintain a nearly constant heat input to the thermoelectric converter in spite of the decay profile of a short lived isotopic heat source. Excess thermal power available early in life, is automatically shorted around the thermoelectric module by way of the heat pipe.

A development model of a gas controlled heat pipe capable of performing the required task was constructed and tested during this program. This heat pipe demonstrated approximately a 6:1 turn down ratio with a 20$^{\circ}$C heat pipe temperature change. By enlarging the gas reservoir, this same heat pipe demonstrated a 4:1 turn down ratio with a 1$^{\circ}$C heat pipe temperature change. With isotopic fuel sources, a 4:1 turn down ratio corresponds to two isotope half lives. Another major aspect of this program was the evaluation and selection of each of the systems components. These included the isotopic fuel source, the thermoelectric converter and the assembly configuration. The choices selected in each of these areas were:

1. Promethium 147 as Isotopic Fuel Source
   (Cobalt 60 was selected as a backup).

2. Bismuth-Telluride Thermoelectric Converter.

3. Single Heat Pipe with Thermoelectrics mounted on accumulator block as development design model.

This SBIR program had three separate areas of work. These areas were:

Task 1 - Survey of Technology Base

Task 2 - Design of Proof of Principle Heat Pipe

Task 3 - Heat Pipe Fabrication and Test

The program was completed on schedule and on budget. A detailed report of all work performed during this program is contained herein.

# 1.0 INTRODUCTION

Present designs for radioisotope thermoelectric generators (RTG's) are based on the use of plutonium as a heat source. While the RTG's operation has proven to be satisfactory, to date they have made use of highly toxic and environmentally sensitive isotopes. If shorter lived, less toxic isotopes could be used, a much greater acceptance of this type of system might be realized. In order to adapt these less toxic isotopes to this use, a technique must be found which can supply a constant power density to the thermoelectric module. A gas controlled heat pipe has this capability. It has the ability to alter the transmitted heat flux characteristics from an isotopic fuel source.

The heat pipe is a sealed heat transfer element. It makes use of vapor heat transfer to carry heat at low temperature loss from an input area where a fluid is evaporated, to an output area where the vapor is condensed, giving up its heat of vaporization. The heat pipe uses capillary action in a porous wick to return the condensed liquid to be re-evaporated (Figure 1).

Heat pipes have several interesting properties. First, and most obvious, is their high effective thermal conductance, which may exceed that of the best solid conductors by factors of 10 or more than 100. The second is their ability to concentrate or disperse thermal heat flux density. This property has become known as heat flux transformation.

An additional advantage comes with the adding of a noncondensible gas reservoir to the heat pipe. This allows for a flattening of the power decay profile from isotopic heat sources with only a slight temperature variation in time. A more detailed explanation of a gas controlled heat pipe is presented in the next section.

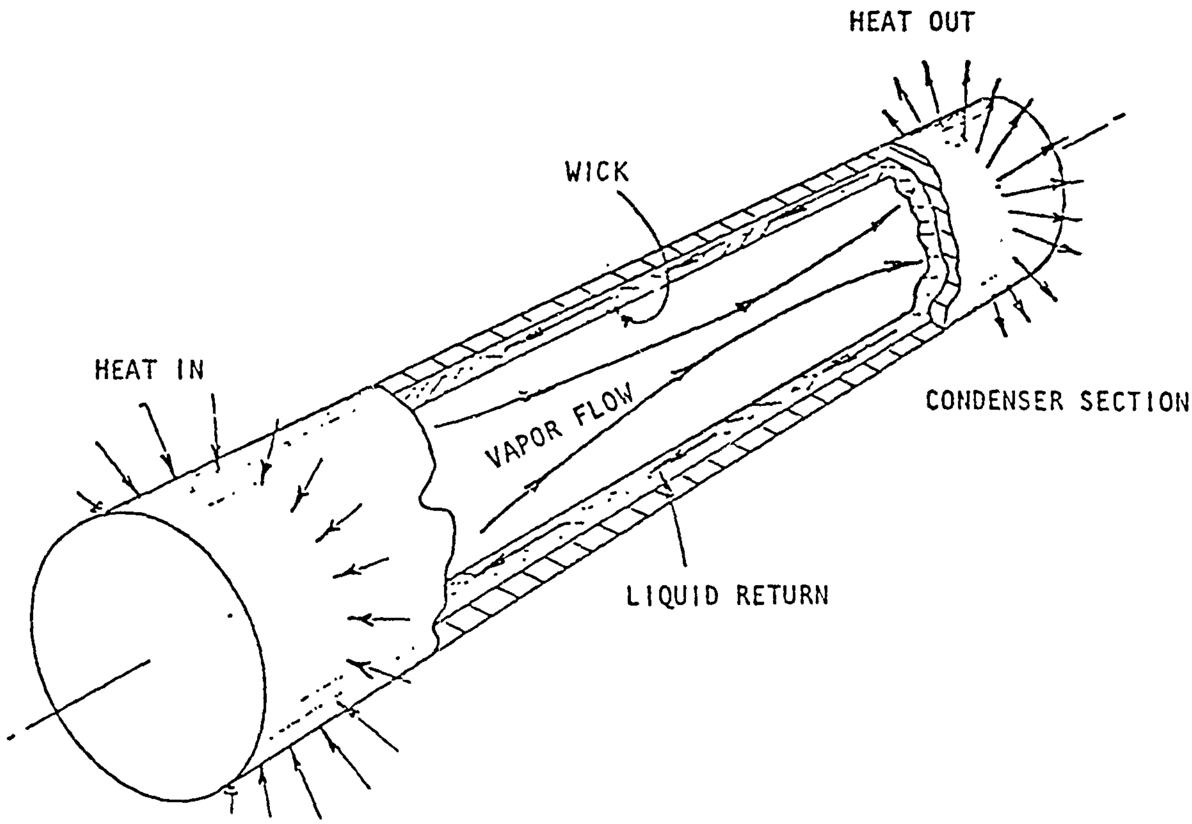


Figure 1. Basic Heat Pipe Schematic

The first use of a gas controlled, or variable conductance heat pipe was at RCA in 1964. Thermacore employees, previously with RCA, were directly involved in this and several other programs which are still state of the are in this area of heat pipe technology.

The objective of the present program was to bring this dormant technology up to date and show the application of heat pipes in RTG assemblies to be feasible. This was done with the development and fabrication of a proof of principle gas controlled heat pipe with a 4:1 turn down ratio, the equivalent of two isotope half lives.

## 1.1 Gas Controlled Heat Pipes

By adding a gas reservoir to the conventional heat pipe shown in Figure 1, it becomes possible to operate a heat pipe at a nearly constant temperature while the rates of power removal and addition are altered. The proof of principle heat pipe constructed in this program physically demonstrated this interesting principle. This section of the report will be used to give a broad based explanation of the components and operation of a gas controlled heat pipe. Figure 2 shows a gas controlled, or variable conductance, heat pipe. The pipe can be divided into four individual components. They are:

1. envelope
2. wick structure
3. working fluid
4. gas reservoir and non condensible control gas

The envelope is the containment vessel for the working fluid. It can be made from many materials, including common metals such as copper or stainless steel, or it can be made from more unusual materials such as nidoium or tungsten. The envelope must be structurally sound, being able to withstand the stresses generated by the internal pressure

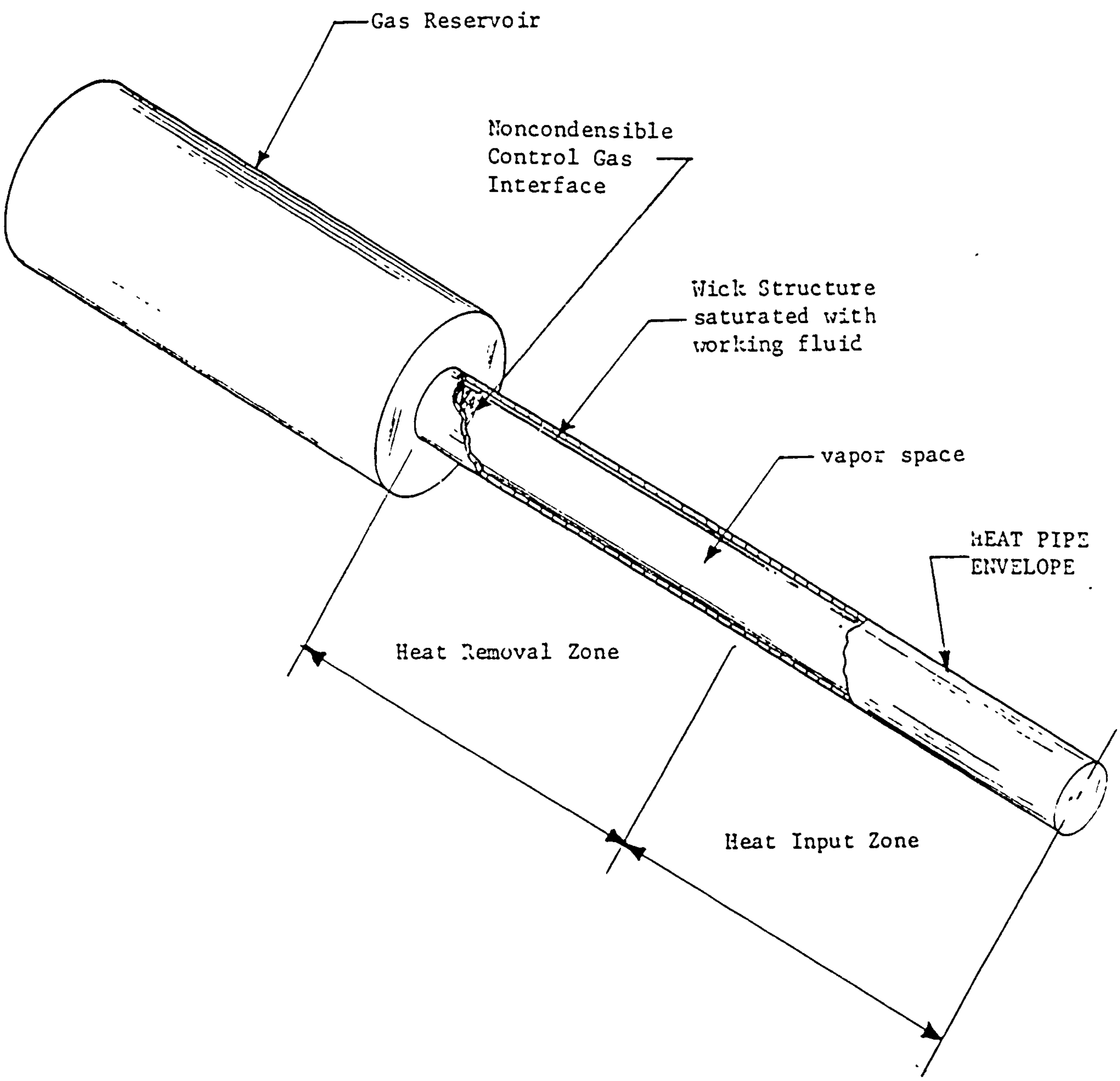


Figure 2 - Gas Controlled Heat Pipe

of the working fluid. it must also be impermeable to the working fluid, preventing diffusion of the fluid through the walls. In addition, the envelope material must be chemically non-reactive with the working fluid. Any reaction between these two would seriously degrade the useful life of the heat pipe.

The wick structure uses capillary pumping for the transport of the working fluid within the heat pipe. Liquid is moved from the condenser section, where the vapor condenses, to the evaporator section where it can again be evaporated. The wick also distributes the working fluid within the evaporator section, allowing evaporation to occur from the entire surface. As with the envelope, the wick material must be chemically non-reactive with the working fluid. Another requirement of the wick structure is that it must be able to supply a sufficient amount of liquid to the evaporator in order to sustain operation. This can be accomplished by varying the configuration and material used in constructing the wick. A sintered powder metal wick, because of its small particle size, provides a much greater capillary pumping capability than a screen wick. The addition of arteries provides low drag fluid return channels to the evaporator, increasing the fluid return capabilities of the wick. Several configurations of wick structures are shown in Figure 3.

The working fluid is used to transport energy within the heat pipe from evaporator to condenser. It must wet the wick and wall material well if a high pumping capability is required. Wetting requires that a cohesive force exists between the molecules in the working fluid and in the wick and wall materials. The degree to which a fluid wets a surface is described by the wetting angle of the fluid on that surface. If the angle is less than $90^{\circ}$, the fluid is said to wet that surface;

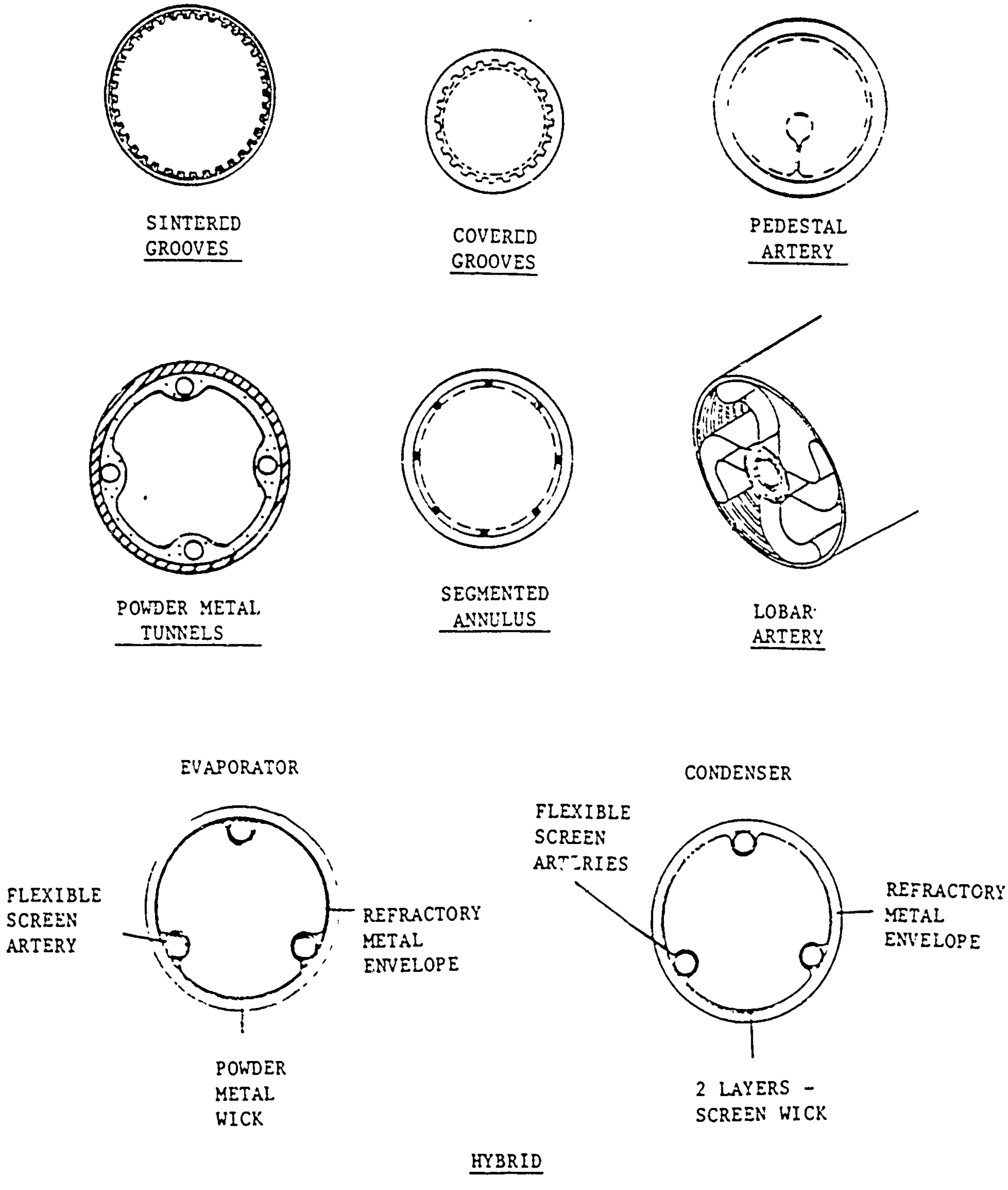


Figure 3 - Wick Configurations

if the angle is greater than 90°, the fluid does not wet that surface. A wetting angle of 0° means the fluid perfectly wets the surface.

The gas reservoir is really an extension of the envelope. The reason it is considered a separate component here is that it is not part of the active vapor space in the heat pipe. Its sole purpose is to contain a non-condensible gas, which will be used to change the power characteristics of the heat pipe. The non-condensible control gas must not react with the working fluid and must not degrade over the operating life of the heat pipe.

Each of these areas was investigated during the design of the proof of principle heat pipe. A detailed explanation of the analysis and conclusions can be found in Section 2.3 (Task 2 - Design of Proof of Principle Heat Pipe).

The operation of a heat pipe is deceptively simple. A liquid is evaporated from the wick structure in the heat input zone and travels to the heat removal zone where the vapor condenses, giving up its heat of vaporization. The control gas in a variable conductance heat pipe allows the heat transfer surface area of the condenser to be changed.

The operation of a heat pipe is governed by a variety of pressure balances. A pressure difference in the vapor space drives the vapor from the evaporator to the condenser. Another pressure difference, this one in the wick structure, drives the fluid from the condenser back to the evaporator. With a gas controlled heat pipe, the control gas is put into the heat pipe at a pressure equal to the vapor pressure of the working fluid during steady state operation. As the power input decreases, the vapor pressure of the working fluid will decrease. This change will cause a movement of the gas interface in the condenser (Figure 4).

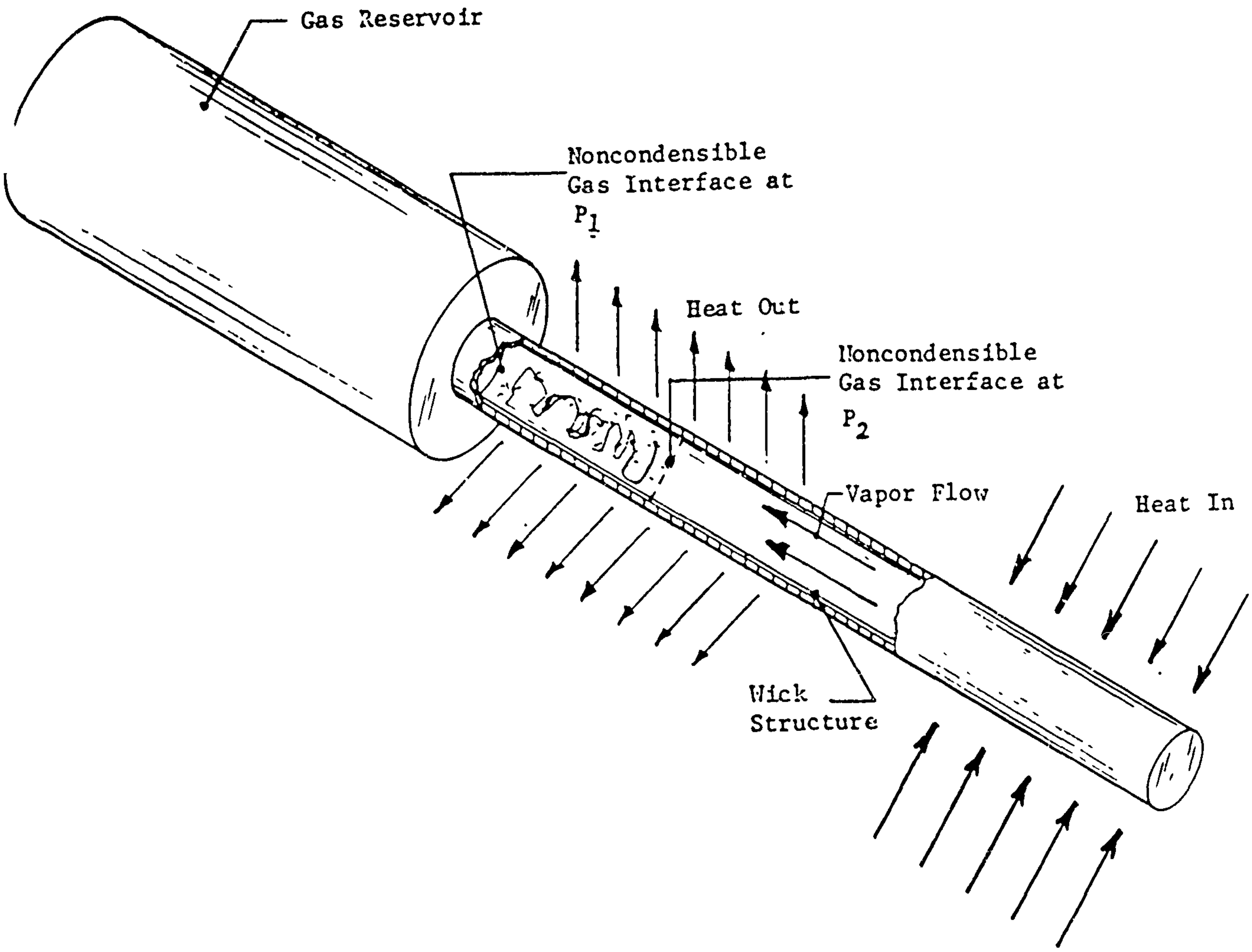


Figure 4. Operation of Gas Controlled Heat Pipe

A heat pipe can be designed so that the gas interface is moved to a specific location by a specific temperature change. This is the principle on which the heat pipe constructed during this program is based.

Referring to Figure 4, a slight decrease in heat source power results in a slight decrease in the evaporator temperature. Through the vapor pressure/temperature relationship of the working fluid, this causes a decrease in the vapor pressure. As a consequesnce, the control gas expands to match the new vapor pressure, moving the gas/vapor interface to a new location. The new interface location establishes a new balance between heat input and heat output. That is, the interface moves until the control gas cuts off vapor access to the excess power radiator to compensate for the reduced input. This action has the effect of maintaining the heat pipe temerature at a nearly constant value over a large power excursion. Heat flow to the thermoelectric module is maintained through several half lives by this method. Operation is fully automatic and requires no external controls or power.

## 2.0 DESCRIPTION OF WORK PERFORMED

### 2.1 Objective and Organization

The objective of this research program was to investigate gas controlled heat pipe technology as a means of flattening the thermal decay profile of low toxicity radioisotopes for RTG service, thereby providing an alternative to the highly toxic isotopes that are presently being used.

Phase I of this program, which was of six months duration, consisted of three separate tasks. They were:

Task 1 - Survey of Technology Base

Task 2 - Design of Proof of Principle heat Pipe

Task 3 - Heat Pipe Fabrication and Test

The work on each of these tasks, as well as conclusions and recommendations, is discussed in detail in the following sections.

### 2.2 Task 1 - Survey of Technology Base

Four areas of technology were investigated so that the requirements for the design of a gas controlled heat pipe could be established. These areas were:

1) Isotopic Fuel Sources
2) Thermoelectric Converters
3) Failure and Safety Mode Considerations
4) Development/Design Concepts

This investigation included both industrial and governmental sources. The information and conclusions from the evaluation are presented in the following sections.

2.2.1 Isotopic Fuel Sources

Radioisotopes decay at widely varying rates. Their associated properties also vary greatly. To date, the most commonly used isotopic heat source for thermoelectrics is plutonium-238. The disadvantage of this isotope is that it is highly toxic and its use introduces a potentially serious environmental hazard. The use of a short-lived isotope can reduce the long-term dangers presented by plutonium-238. Preliminary discussions with Battelle Northwest yielded a rather large selection of isotopes which could be used as heat sources (Table 1)[1].

Requirements were defined which provided a means of evaluation for the screening of the isotopes. The requirements were:

1) Half-life of Compound - the half life should be long enough to operate with a reasonable turn down ratio while using a relatively short-lived isotope with respect to what is currently being used. A minimum half life of one year was selected for this evaluation.

2) Melting Point of Compound - the melting point must be above the operating temperature of the system. This temperature is set by the requirements of the thermoelectrics, with a minimum melting point temperature above 1500°C.

3) Compound Power Density - the compound power density must be high so that the required capsule volume is not excessively great.

4) Isotope Toxicity - the levels and types of radiation emitted by a compound must, respectively, be low and present as

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[1]"Characteristics of Radiostopic Heat Sources," Battelle Pacific Northwest Laboratories, Richland, Washington, 1976.

TABLE 1

# CHARACTERISTICS OF RADIOISOTOPIC HEAT SOURCES

| | $^{60}$Co | $^{90}$Sr | $^{106}$Ru | $^{137}$Cs | $^{144}$Ce | $^{147}$Pm | $^{170}$Tm | $^{210}$Po | $^{238}$Pu | $^{242}$Cm | $^{244}$Cm |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1. Watts/Gram (100% Basis) | 17 8 | 0.933 | 33.1 | 0 416 | 25 6 | 0 333 | 13 6 | 144 | 0 56 | 120 | 2 84 |
| 2. Half-Life Years | 5.24 | 28 6 | 1 0 | 30 | 0 78 | 2 62 | 0 35 | 0 38 | 87.8 | 0 45 | 18 1 |
| 3 Curies/Gram (100% Basis) | 1141 | 139 | 3394 | 87 | 3180 | 928 | 6048 | 4500 | 17 | 3320 | 81 |
| 4. Curies/Watt | 64 2 | 149 | 102 | 209 | 124 | 2786 | 445 | 31 | 30 | 28 | 29 |
| 5. Estimated Isotopic Purity % | 10[a] | 61[b] | 3 6[b] | 45[b] | 5 5[b] | 95[b] | 10[a] | 95 | 80 | 90 | 95 |
| 6. Compound Form | Metal | $SrTiO_3$ | Metal | CsCl | $Ce_2O_3$ | $Pm_2O_3$ | $Tm_2O_3$ | Metal | $PuO_2$ | $Cm_2O_3$ | $Cm_2O_3$ |
| 7 Melting Point of Compound °C | 1495 | 2040 | 2310 | 645 | 2190 | 2320 | 2375 | 254 | 2232 | 2230 | 2230 |
| 8 Active Isotope in Compound[c] | 10 | 27 0 | 3 6 | 34 5 | 4 23 | 82 | 8 8 | 95 | 70 | 75 | 83 |
| 9 Watts/Gram Compound | 1 78 | 0 252 | 1 19 | 0 143 | 1 08 | 0 273 | 1 2 | 137 | 0 39 | 90 | 2.35 |
| 10 Density of Compound, g/cm³, actual or 90% TD | 8 8 | 5 03 | 12 4 | 3 8 | 6 9 | 6 6 | 8 0 | 9 2 | 10 3 | 11 7 | 11 7 |
| 11 Power Density, W/cm³ Compound | 15 7 | 1 27 | 14 8 | 0 545 | 7 47 | 1 8 | 9 6 | 1260[d] | 4 0[d] | 1053[d] | 27 5[d] |
| 12. Dimension of capsule for 50 $W_t$ cm[e] | 2 2 | 4 3 | 2 2 | 5 5 | 2 6 | 3 9 | 2 5 | 1 7 | 3 1 | 1 7 | 1 9 |
| 13. Availability | Avail | Avail | Poten Avail | Avail | Avail | Avail | Avail. | Avail. | Avail. | Poten Avail | Avail. |
| 14 Major Types of Radiation* | γ β | β X γ | γ β X | β γ X | γ β X | β | β X | α X | α | α n | α n |
| 15. Spontaneous Fission Half-Life, Yr | — | — | — | — | — | — | — | — | 4 9x10¹⁰ | 7 2x10⁶ | 1 4x10⁷ |
| 16 Shielding Required[f] (inches of lead) [MeV of significant β or γ] | Heavy (9 5) [1 33γ] | Heavy (6) [2 26β] | Heavy (9) [3 35β] | Heavy (4 6) [1 17β 0 67γ] | Heavy (10 2) [2 98β 2 18γ] | Minor (1) [0 23β] | Moderate (2 5) [0 97β] | Minor (1) [0 8γ] | Minor (0 1) [0 04γ] | Minor (0 4) [0 04γ] | Moderate (2) [0 04γ] |
| 17 Bio Haz, MPC, Total Body, Air, Sol Continuous Exposure — Ci/m³ | 10⁷ | 3x10¹⁰ | 3x10⁷ | 2x10³ | 10⁸ | 2x10⁹ | 7x10⁸ | 2x10⁹ | 5x10⁻¹² | 2x10⁻¹⁰ | 9x10⁻¹² |
| W/m³ | 1 6x10⁹ | 2x10¹² | 3x10⁹ | 9 6x10¹¹ | 8x10⁻¹¹ | 7 2x10¹³ | 1 6x10¹⁰ | 6 5x10⁻¹¹ | 1.7x10¹⁴ | 7 1x10⁻¹² | 3 1x10⁻¹³ |
| g/m³ | 8 8x10¹¹ | 2 2x10⁻¹² | 8 8x10⁻¹¹ | 2 3x10¹⁰ | 3 2x10¹² | 2 2x10⁻¹² | 1 2x10¹¹ | 4 5x10⁻¹³ | 2 9x10⁻¹³ | 6.0x10¹⁴ | 1.1x10⁻¹³ |
| 18 Estimated Future Price, $/g(Pure), [Present Price] | 285 [456] | 20[g] [14] | 120[g] | 10[g] [8 70] | 50[g] | 75[g] [186] | 136 | 2800[h] | 300 | 2000[i] | 185 |
| 19 Estimated Future Price $/W | 16 | 22 | 5 | 24 | 2 | 220 | 10 | 20 | 540 | 17 | 65 |
| 20 Total $kWh_t$ released/ initial gram over a mission period of (yr). | 582 (5) | 73 (10) | 209 (1) | 33 (10) | 150 (1) | 6 (3) | 33 (0 4) | 350 (0 4) | 47 (10) | 310 (0 4) | 209 (10) |
| 21 Minimum Cost, $/$kWh_t$[j] for (yr) mission | 15 | 6 5 | 23 | 7 | 9 5 | 320 | 165 | 235 | 133 | 240 | 22 |
| 22 Grams per ton (at 25000 MWD/t) Recoverable from Reactor Fuel: | — | 360 | 53 | 835 | 89 | 60 | — | — | (345Np) | — | 9 6(204 for Pu recycle) |
| 23 Production in Power Reactors kg/1000 MWY. | | 18 | 2 1 | 42 | 5 3 | 2 9 | — | — | (18Np) | — | 0 5 (10 for Pu recycle) |
| 24 Est Availability in 1980, $kW_t$[m] | MWs | 1100 | 6000 | 1100 | 10,000 | 54 | MWs | MWs | 232[k] | — | 119 185[l] |

little hazard as possible in order to meet the demands set by the program objective.

5) Compound Availability - the selected isotope must be available at levels which can supply full scale development.

6) Compound Cost - the cost, when coupled with the advantage of using an isotope with a lower toxicity, must show such a system to be feasible when compared with what presently exists.

Combining and analyzing the information that was provided by several sources, the field of possible isotopic heat sources was narrowed. The isotopes that presented serious toxicity problems, such as Strontium 90, Ruthenium 106, Cerium 144, Curium 244, and Plutonium 238 were immediately eliminated. Two other candidate isotopes, Polonium 210 and Cesium 137, were eliminated as possibilities because their melting points were below the temperatures at which thermoelectrics operate. Calculations were performed on the remaining candidate isotopes so that they could be compared on the basis of system life and the volume each would occupy when used as a heat source for a thermoelectric generator (Appendix A).

In order to perform these calculations, two assumptions were made. First, when working with short-lived isotopes such as those being considered, a reasonable length of time should be included so that the isotope capsule can be stored after filling but before deployment. After discussions with DARPA, a time period of two isotope half lives with a minimum of one year was selected as a sufficient length of time to satisfy this requirement. This system life will be used throughout the development program, including the heat pipe design. Second, a 58 watt thermal output from the isotope capsule will be required at the end of the system's life. This is the input required by a Bismuth-Telluride thermoelectric module which will be capable of supplying one watt of electricity. The Bismuth-Telluride thermoelectric module is used here purely as a reference on which the volumes of the isotope capsules can be calculated and compared. An evaluation of the various types of thermoelectrics that are available will be covered in Section 2.2.2 of this report.

TABLE 2. SELECTED CANDIDATE ISOTOPES

| Isotope | System Life (years) | Capsule Volume ($cm^3$) | System Life Capsule Volume Ratio |
|---|---|---|---|
| $^{60}Co$ | 10.5 | 14.9 | 0.71 |
| $^{147}Pm$ | 5.24 | 129.3 | 0.041 |
| $^{170}Tm$ | 0.7 | 24.2 | 0.029 |
| $^{242}Cm$ | 0.9 | 2.3 | 0.39 |

The results of the isotope calculations can be seen in Table 2. The system life to isotope capsule volume ratio is one means of comparison between the isotope's properties and the other pertinent issues, such as acceptable radiation levels and isotope availability. If all other conditions are held constant, the larger this ratio, the more suitable the isotope will be as a heat source. As can be seen, Thulium-170 and Curium-242 have small life to volume ratios. Based on this ratio and the fact that there appears to be no apparent advantage in their selection, these two isotopes were eliminated as possible choices. Promethium-147 offers the most promise of all short-lived isotopes that have been evaluated as potential heat sources for RTG service. While Promethium-147 has a small life to volume ratio, the low toxicity and light shielding require ments indicate the advantages of using this isotope are far greater than its single disadvantage of not having a large power density. Based on these facts, Promethium-147 was selected as the primary reference isotope for this program.

Cobalt-60 was also given serious consideration during this evaluation. Although it is more toxic than Promethium-147 and requires approximately nine times the shielding for the same level of radiation protection, it does have several advantages. Because of its extensive use in the field of medicine it has achieved a rather high level of public acceptance. In addition, a large collection of information is available on its safe handling and processing. Because this is an artificial isotope that can be manufactured whenever needed, the question of availability and the neeed for stockpiling are eliminated. Based on this reasoning, Cobalt-60 was selected as a backup isotope if for some reason Promethium-147 is not feasible.

Although Promethium-147 is a byproduct of other nuclear programs, it is believed that sufficient quantities are available to warrant its use as an isotopic heat source for thermoelectric generators. The Department of Energy has supplied estimates for current and future inventories of Promethium-147 at their Hanford and Savanna River Plant facilities. This data is listed in Table 3.[2] The results of the isotopic heat source evaluation are as follows:

Primary Isotope Choice: Promethium-147

Secondary Isotope Choice: Cobalt-60

2.2.2 Thermoelectric Converters

Thermoelectric converters provide a means of converting heat directly into electricity. Presently available thermoelectric modules are made

[2]"Department of Energy Plan For Recovery and Utilization of Nuclear Byproducts from Defense Wastes," Document DOE/DP-0013, August 1983.

TABLE 3. ESTIMATED INVENTORIES OF PROMETHIUM-147 FROM VARIOUS SOURCES

| | | Effective Inventory Date | kg | Watts (t) |
|---|---|---|---|---|
| Hanford | - Current wastes | 1/83 | 9 | $3 \times 10^3$ |
| | - Future wastes | 1/91 | 39 | $1.3 \times 10^4$ |
| Savannah | - Current wastes | 1/83 | 120 | $3.9 \times 10^4$ |
| Plant | - Future wastes | 1/2001 | 206 | $6.9 \times 10^4$ |
| Commercial | - Accumulated through 1981 | 1/82 | 460 | $1.1 \times 10^5$ |
| | - Accumulated through 2020[a] | 1/2021 | 3,300 | $1.1 \times 10^6$ |

[a] Assumes no interim processing.

from a variety of materials and operate through a large temperature range. This evaluation considers Bismuth/Telluride, Lead/Telluride, and Silicon/Germanium thermoelectric modules. The characteristics of each of these are presented in Table 4.

In each case, because of the small electrical output requirement (1W), a surface area less than 6.5 $cm^2$ is needed. Coupling this fact with the close relationship between their efficiencies, it can easily be seen that the only characteristic which varies greatly between the available thermoelectrics is their hot shoe temperature requirement.

The criterion which guided the selection of a reference thermoelectric material was a low hot shoe temperature requirement in comparison with other available thermoelectrics. The entire isotope-heat pipe-thermoelectric generator system being designed is only one component in a larger system of electronic hardware. The overall system design must consider the effects each component will inflict on the entire system. From this, the conclusion was made that the electronics would be least affected by providing the lowest possible environment temperature. Since the hot shoe temperature for Bismuth/Telluride is the lowest available, being 270°C lower than the next possible choice, its selection was considered the most feasible.

The results of the thermoelectric generator evaluation are:

Primary Thermoelectric Generator Choice:

BISMUTH/TELLURIDE

with an operating temperature of 270°C.

### 2.2.3 Safety/Failure Mode Considerations

Any system which uses a radioactive element as a component must make safety one of the primary concerns. In an RTG system such as the

TABLE 4. THERMOELECTRIC CONVERTER CHARACTERISTICS

Bismuth Telluride

Operating Temperature (Hot Shoe): 260 - 270°C

Device Efficiency: 4 1/2 - 5 1/2%

System Thermal Efficiency: 60 - 85%

Power Converter Efficiency: 80 - 85%

Thermal Input for 1 We: 58W - 32 W

Required Power Densities: 8.99 W/cm$^2$ - 4.96 W/cm$^2$

Forms: Elements, Modules

Lead Telluride

Operating Temperature: 538°C

Device Efficiency: 8 1/2 - 9 1/2%

System Thermal Efficiency: 60 - 85%

Power Converter Efficiency: 80 - 85%

Thermal input for 1 We: 31W - 18W

Required Power Densities: 4.81 W/cm$^2$ - 2.79 W/cm$^2$

Forms: Elements

Silicon/Germanium

Operating Temperature: 1,000°C

Device Efficiency: 9.1%

Thermal Efficiency: 80 - 90%

Converter Efficiency: 80 - 85%

Thermal Input for 1 We: 21.5W - 18W

Required Power Densities: 3.33 W/cm$^2$ - 2.79 W/cm$^2$

Forms: Multi-Couple Module

one being considered, any advantage which could be gained by using a less toxic isotope would be lost if this new system design could not provide adequate safeguards for the isotope.

In addition to providing a safe environment for the isotope capsule, this design must be as reliable as the RTG systems that are presently being used. The only primary component which is used in the new design, but not in the old, is the heat pipe. Therefore, this design must show the heat pipe will not in any way reduce the reliability of the system and most importantly, the heat pipe must provide a safe environment for the isotope capsule.

The following section will show the progression of concepts which were devised to overcome any difficulties which were foreseen. In each case, system safety and reliability were analyzed and the concept adapted to serve each to the fullest possible advantage. The final design described is the culmination of numerous concepts into a safe, reliable, and feasible design.

2.2.4 Development of Design Concepts

In the design of this system several configurations were considered and evaluated on the level of safe and reliable operation that each would provide. While a 100% failure-proof design is impossible, the goal of this design was to provide as many alternatives to an uncontrolled failure as possible. Also of importance is the individual component's feasibility and fabricability which must be kept in mind when making any design decisions. The final concept described is the incorporation of the advantages that each concept demonstrated by the individual models.

The location of the isotope capsule with respect to the heat pipe was the first consideration addressed. Figure 5 illustrates two of the possible designs.

The first sketch, Figure 5(a), represents an annular isotope capsule surrounding a conventional heat pipe. This configuration offers the advantages of an easily fabricated heat pipe and a means of failure protection for the isotope capsule. In the event of a heat pipe failure, the possibility exists for the isotope capsule to radiation cool itself, losing its heat to the surroundings without melting the capsule. The disadvantage of this configuration is the annular isotope capsule design. Accordingly, since heat is being removed from the inner radius of the isotope capsule, a temperature differential exists between the inner and outer radius. Because the rate of thermal expansion is a function of temperature, stresses will be created in the capsule shell. These stresses could result in a structural failure of the capsule.

In the second concept shown, Figure 5(b), the heat pipe is slightly more difficult to fabricate. The isotope capsule is located in a reentrant cavity within the heat pipe. The advantage to doing this concerns the thermal losses from the isotope capsule. When using short-lived isotopes, such as Promethium which have very small power densities, the capsule volume becomes an important consideration. In order to keep the capsule volume as small as possible, the thermal losses from the system must also be kept very small. By surrounding the isotope capsule with the evaporator of the heat pipe, the thermal losses from the capsule are reduced to virtually zero. Another advantage to the reenterant design is the isotope's capsule quick and easy fabrication. This is an important characteristic if the system is stored for any length of time because,

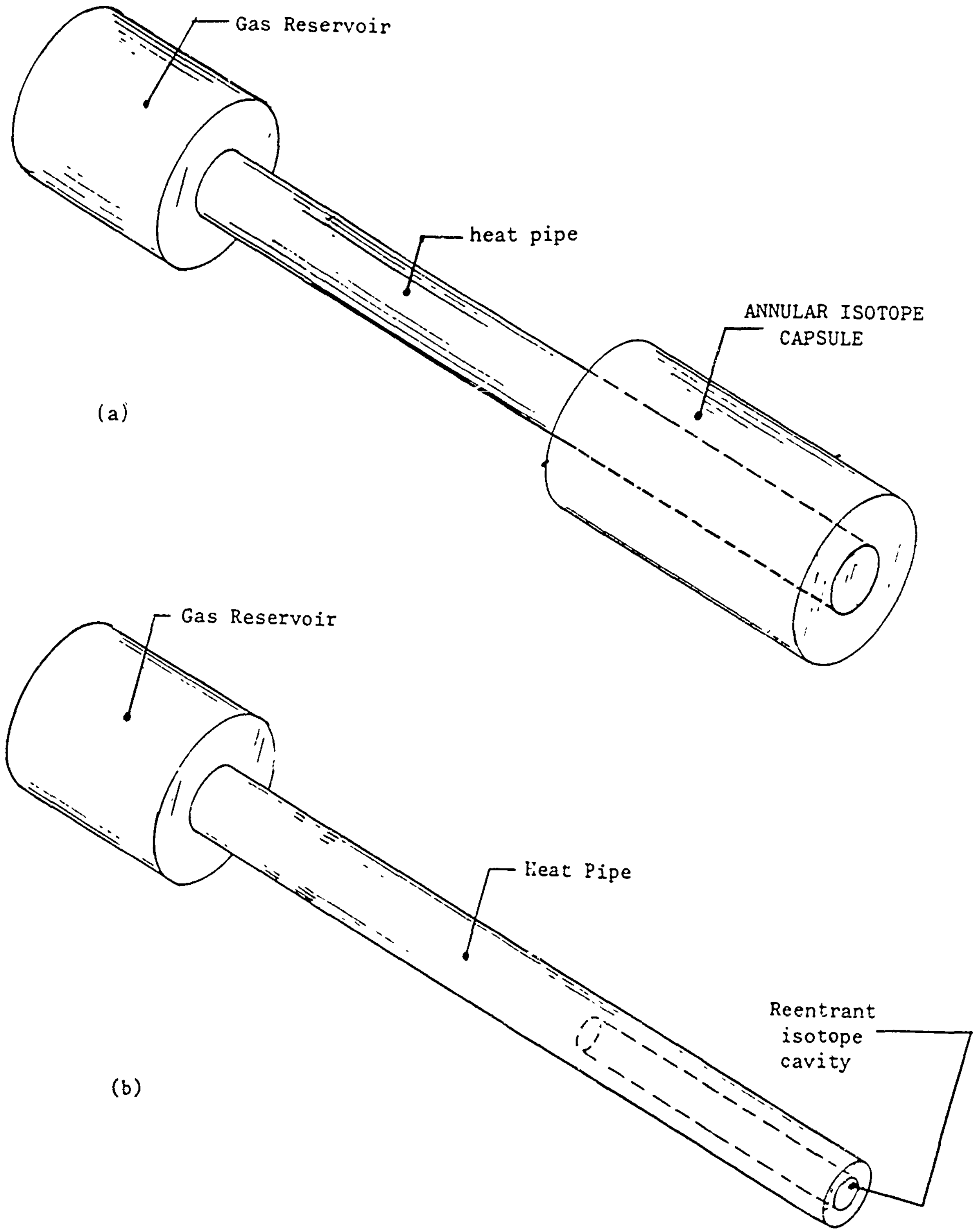



Figure 5 - Isotope Capsule Location

(a) Heat Pipe with Annular Isotope Capsule

(b) Heat Pipe with Cylindrical Isotope Capsule while the heat pipe assembly has a rather long-shelf life, the isotope capsules do not, and the stock will continually have to be rotated and replaced.

The heat pipe reliability/safety consideration was brought to our attention by a thermoelectric manufacturer. He was unsure of the feasibility of locating the thermoelectric module directly on the heat pipe, believing instead that the thermoelectric module should be mounted on one end of the isotope capsule with the heat pipe on the other end of the capsule being used to dissipate the excess heat (Figure 6). When the isotope and thermoelectric characteristics were reviewed in analyzing this suggestion, it was found that this configuration was not possible. The power density requirements of the thermoelectrics are greater than the power density output from an isotope capsule fueled with a short-lived isotope, such as Promethium-147. A 2.54 cm diameter cylinder of 147 Pm has a power density of 1.1 $W/cm^2$, while a Bismuth Telluride thermoelectric requires between 5 $W/cm^2$ and 9 $W/cm^2$. This difficulty is unique to the short-lived isotopes, such as 147 Pm, and therefore has not previously been a problem that thermoelectric generator manufacturers have had to contend with.

Discussions with Teledyne Energy Systems, a thermoelectric manufacturer, introduced an important point; the system's safe operation in the event of a heat pipe failure. Previously, a single heat pipe was being considered to power a single thermoelectric module. What was not acceptable in this design was the fact that if the heat pipe were to fail, the isotope capsule could melt down. Figure 7 illustrates two concepts which provide heat removal by a redundancy of heat pipes. This redundancy allows for safe operation by the fact that, if a single or even two heat pipes

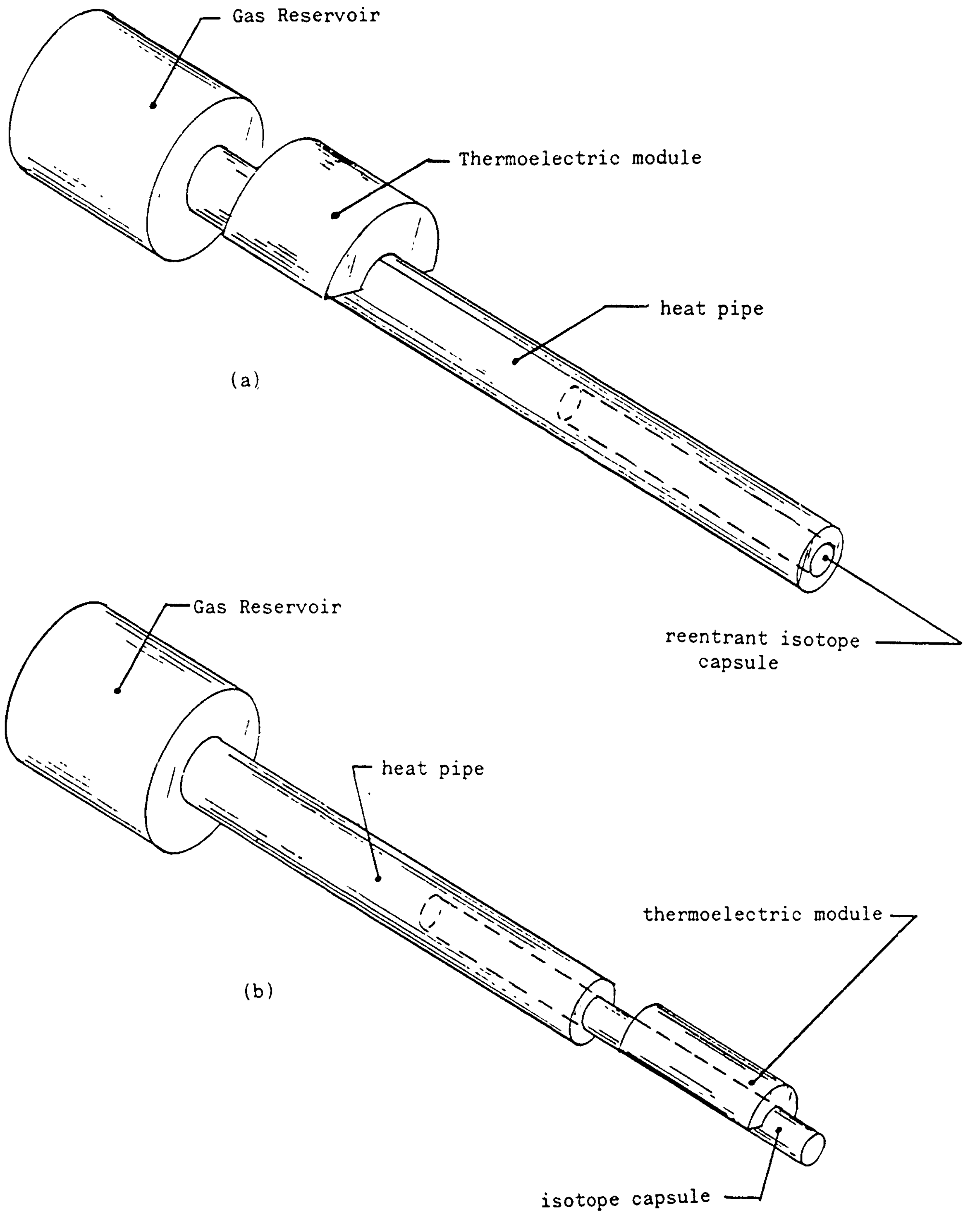



Figure 6 - Thermoelectric Module Location

(a) on envelope

(b) on isotope capsule

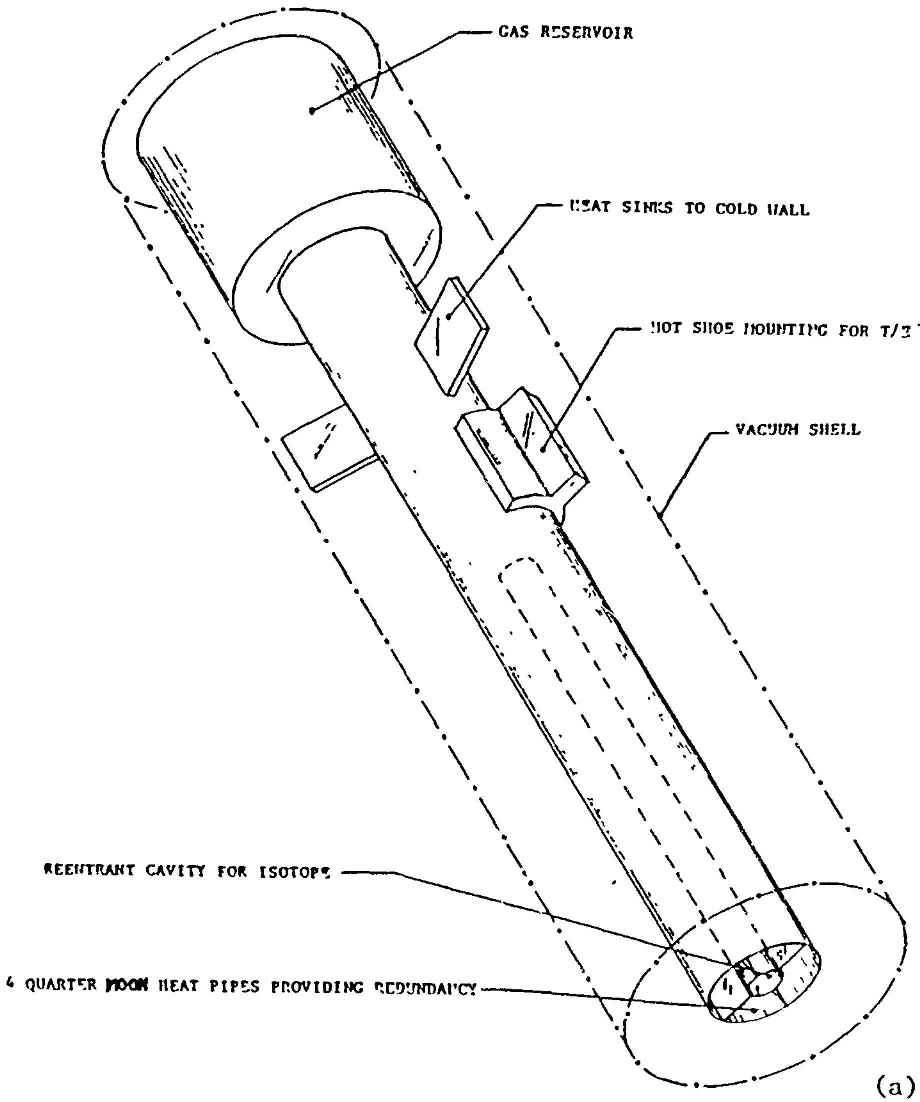
were to fail, the system could be cooled by the remaining heat pipes, reducing the possibility of the capsule melting. In fact, this design provided the ability for a single heat pipe to carry and dissipate the entire output from the isotope capsule.

Figure 7(a) illustrates one way of providing this redundancy. The concept calls for a heat pipe with a reentrant cavity in one end. The isotope capsule would be inserted in this cavity. Internally, the pipe was divided into four individual quarter-moon-shaped heat pipes. By doing this, all four heat pipes wold have to fail before the capsule could melt. While this system could operate with a reduced chance of a capsule melting, the heat pipe's fabricability was unnecessarily difficult. Because of the numerous long welds along critical joints, the possibility of failure along these welds becomes an important concern in that the chance of a heat pipe failure was substantially increased.

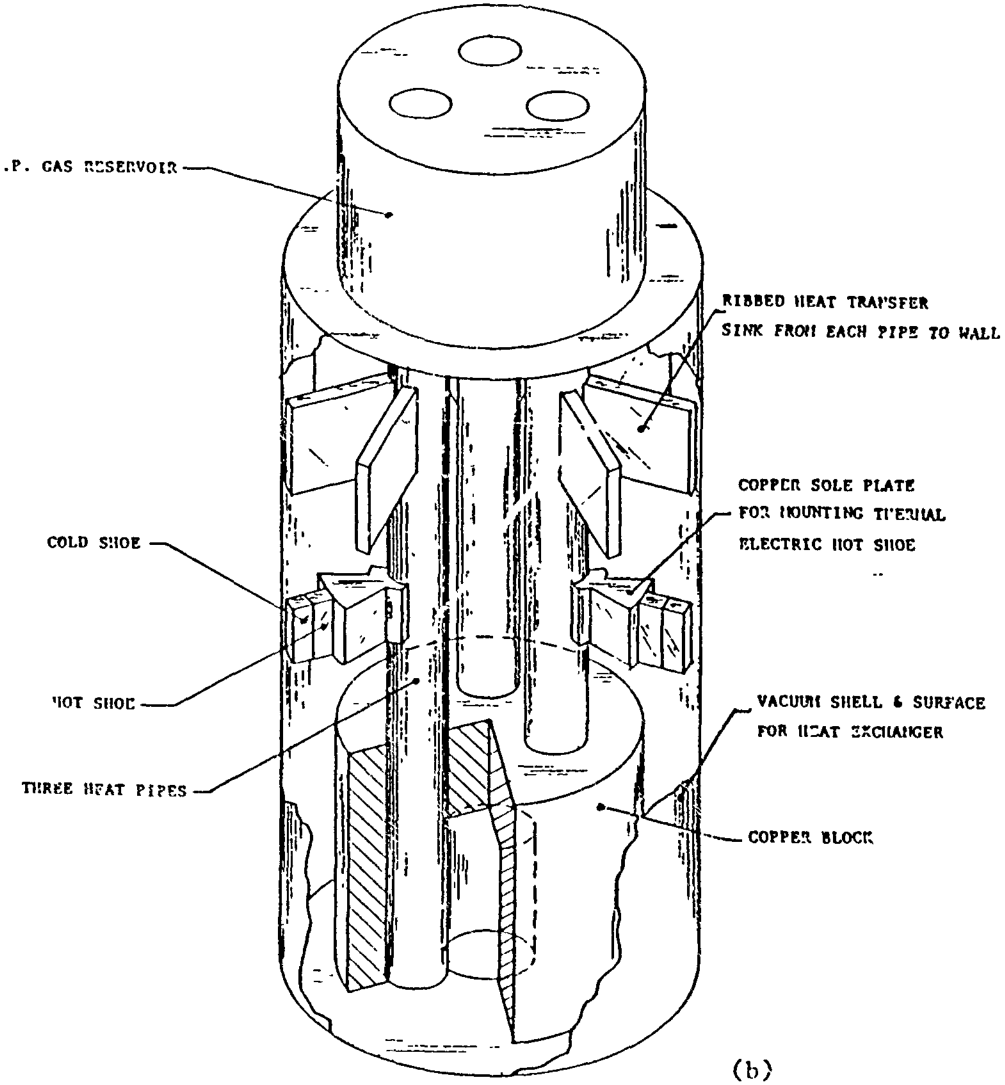
A system which combines heat pipe redundancy with ease of fabrication is shown in Figure 7(b). This configuration incorporates many of the advantages previously discussed. Redundancy is provided by three individual heat pipes. Each heat pipe is sized so that if two of the three were to fail, the one remaining heat pipe will operate at the specifications set for the system, with the thermoelectrics on the pipe supplying one watt of electricity. In normal operation, each heat pipe would supply enough energy to a thermoelectric module for an output of one-third of a watt of electricity while also dissipating one-third of the excess power.

All three heat pipes in this system would be powered by a single isotope capsule located in a cavity in the solid copper cylinder shown in the Figure. Surrounding the capsule, three holes would be bored

Figure 7 - Selected Concepts Providing Heat Pipe Redundancy




(a)




(b)

in the cylinder 120 degrees apart. Into each hole the evaporator of a heat pipe would be inserted. This design is very similar to the concept in which the capsule was located within a reentrant cavity in the heat pipe evaporator.

If for some unexpected reason all three heat pipes were to fail, the large copper cylinder at the base adds still another degree of safety to the system. During normal operation the copper will be surrounded by several layers of reflective insulating foil, limiting the cylinder's thermal losses to a minimum, with the heat pipes carrying away all the power generated in the isotope capsule. If all the heat pipes were to fail and stop carrying power, the temperature of the copper cylinder would rise. The melting point of the foil is well below the melting point of copper so that at some elevated temperature the reflective foils would melt away, allowing the copper block to radiate the heat away to the surroundings from its new elevated temperature. In doing this, the capsule would be cooled, preventing its melting.

The drawback to a system such as this one is that it is expensive and is also excessively heavy. In order to reduce cost and weight, a system containing a single heat pipe was considered (Figure 8). To avoid the threat of the capsule overheating because of a heat pipe failure, the thermoelectric module was mounted directly on the tungsten accumulator block. This was done because the block sufficiently concentrates the thermal power density of the isotope, providing the level needed by the thermoelectrics. Thus, while the thermoelectrics cannot be mounted on the isotope, the use of an intermediate accumulator solves this problem, with the heat pipe used solely to dissipate excess power. Also by doing this only one thermoelectric module is needed instead of the three previously discussed. As with the previously described concept, the block would be surrounded by an insulating foil which would melt away should a heat pipe failure occur. The radiation cooling equilibrium temperature of the block would be much higher than what an operating Bismuth-Telluride thermoelectric module could tolerate, but it does provide a method to prevent overheating of the isotope capsule. This system, the incorporation of many of the advantage discussed in the previous concepts, was considered to be the most feasible of all those considered. This model was, therefore, selected as the reference design configuration of this program.

## 2.3 Task 2 - Design of Proof of Principle Heat Pipe

Task 2 was comprised of the design analysis and calculations for a gas controlled heat pipe which can demonstrate the ability to flatten the power profile from a decaying isotope. The work was divided into several separate areas. These areas were:

- Design Parameters
- Material Selection
- Operating Limitations
- Heat Pipe and Reservoir Sizing

### 2.3.1 Design Parameters

The parameters used in designing the heat pipe are listed in Table 5. They were selected as the results of the evaluation which was described in Task 5 of this report and, in turn, are using the design model previously described.

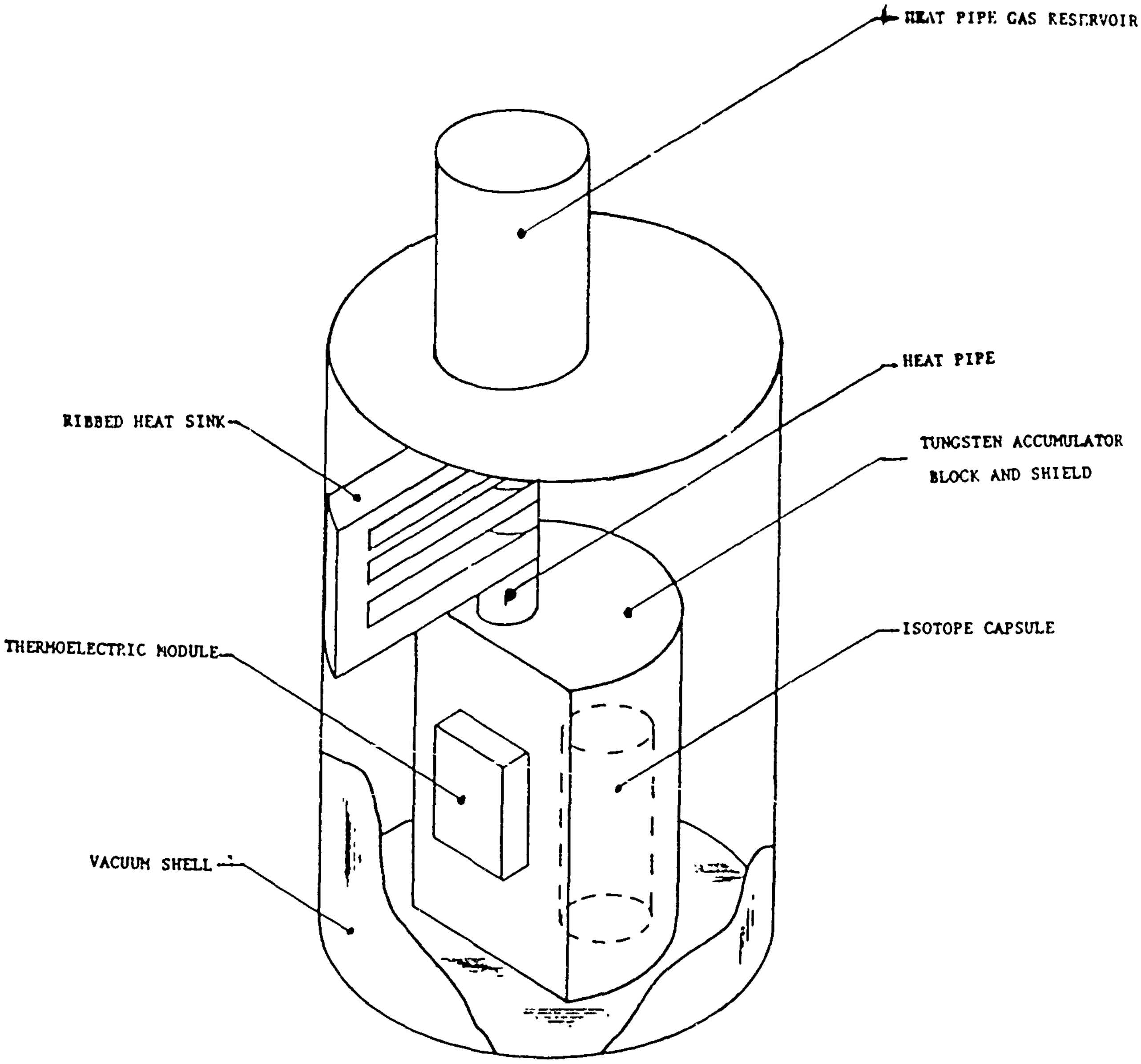



Figure 8. Single Heat Pipe Development Model

TABLE 5. HEAT PIPE DESIGN PARAMETERS

| | |
|---|---|
| Heat Pipe Outside Diameter | 1/2" |
| Heat Pipe Length | 19" |
| Heat Pipe Operating Temp. | 300°C |
| Maximum Heat Load | 250W |
| Minimum Heat Load | 19W |

The overall objective of this task was to design a gas controlled heat pipe which can meet the following requirements:

TABLE 6. PREDICTED OPERATING CONDITIONS

| Case | Power Input(W) | Power To T/E(W) | Power Out Excess(w) | Movement of Non-Condensible Gas Interface(in) | ΔT From 300°C |
|---|---|---|---|---|---|
| 1 | 233 | 58 | 175 | 0 | 0 |
| 2 | 58 | 58 | 0 | 2.75 | 5°C |

Cases 1 and 2 demonstrate the heat pipe operation at the beginning and end of the system's life, respectively. In these two cases, the heat pipe is supplying enough power for an output of one watt of electricity. This would simulate the model described in Task 1, with a single heat pipe being used. By demonstrating these two cases it is felt that any combination of operating conditions between these extremes can be met.

### 2.3.2 Material Selection

Care must be exercised in material selection with respect to the material's chemical compatibilities with one another. It can clearly be seen that the heat pipe envelope and gas reservoir, wick, and working fluid must all be chemically nonreactive in order to assure long life and successful operation.

2.3.2.1 Working Fluid

The heat pipe's operating temperature is the governing factor in working fluid selection. Either directly or indirectly, it controls many of the following requirements used in selecting a working fluid:

1) Good thermal stability
2) Vapor pressure must not be too high or too low in the operating temperature range.
3) High latent heat of vaporization
4) High thermal conductivity
5) Low liquid and vapor viscosities
6) High surface tension

Several heat pipe working fluids were considered for this application. In the temperature range specified by the requirements of the Bismuth-Telluride thermoelectric module (275-300°C) the possible working fluids included Dowtherm A, Rubidium, Cesium, and Mercury.

When evaluating these choices, based on the previously stated requirements, the field was narrowed. Dowtherm A is not thermally stable and, in fact, will begin to decompose within the design lifetime of the system. Of the remaining choices, mercury was concluded as being the best suited working fluid for this application because of its advantageous vapor pressure, high latent heat of vaporization, high surface tension, and compatibility with potential envelope materials.

2.3.2.2. Envelope Material

The heat pipe wall material, including the gas reservoir, must be compatible with mercury, the selected working fluid. Areas of concern during this material selection process included solubility of the working fluid in the envelope walls and embrittlement of the wall material by the working fluid. Stainless steel and low carbon steel were considered as materials of construction. After analysis, low carbon steel (1018 Grade) was selected as the envelope material.

2.3.2.3. Wick Material

The wick is the most important component in a heat pipe, since to a large extent, it controls the heat flux and startup characteristics of a given heat pipe design. The primary function of the wick is to return the working fluid from the condenser section of the heat pipe to the evaporator. A second function the wick serves is to spread the working fluid evenly over the entire evaporative surface.

As with the wall material, compatibility of the wick with the working fluid is a serious concern. Iron is compatible with mercury and was selected as the wick material.

2.3.3. Operating Limits

Four heat pipe operating limits have been defined. They are: (1) capillary wicking height limit, (2) sonic limit, (3) vapor shear limit, and (4) evaporation limit. Because this design has very low heat flux requirements, it is not expected that the heat pipe will be operating near its capabilities.

2.3.3.1 Capillary Wicking Height Limit

The capillary wicking height is a limit in that, in vertical operation, the working fluid must be able to wick at least the height of the isotope capsule. This is necessary to prevent overheating both in the isotope capsule and in the wick structure during startup.

Once the isotope capsule is inserted into the system, both its temperature and the heat pipe's temeperature will rise. The wick structure of the heat pipe must be saturated with working fluid so that it can remove this heat input. If the heat is not removed, the risk exists of the capsule overheating to the point where it could melt down, or hot spots forming in the wick structure, thereby preventing the fluid from wetting that area of the wick.

The following relationship is a balance of the forces acting on the fluid in the wick structure. It equates the force of gravity working against the fluid rise and the capillary pressure capabilities of the wick structure.

$$\rho g h = \frac{2\sigma\cos\theta}{r_c} \qquad \text{(Eq. 1)}$$

$\sigma$ = surface tension of working fluid

$\rho$ = density of working fluid

g = acceleration due to gravity

h = required wicking height

$\theta$ = wetting angle of working fluid on wick material

$r_c$ = capillary radius of wick material

If the working fluid's properties, the degree to which the fluid wets the wick and the height the fluid must wick, are known, the required capillary pore radius can easily be calculated. This calculation is performed in Appendix B with the result presented below. The $57^{\circ}$ wetting angle used is an experimental result for mercury on stainless steel. At present, no data is available for mercury on low carbon steel, but wetting at least equivalent to $57^{\circ}$ is assumed.

$$r_c = 0.0015 \text{ cm} \qquad \text{(Eq. 2)}$$

A capillary pore radius of this order is in the class of powder metals. Therefore, from this it was concluded that an iron powder metal wick structure would be used in constructing the heat pipe.

2.3.3.2 Sonic Limit

In a heat pipe, the vapor stream accelerates and decelerates because of vapor addition in the evaporator and vapor removal in the condenser. This type of flow can be described as variable mass flow through a constant area. As in a converging/diverging nozzle, with constant mass flow through a varying area, a heat pipe vapor flow can reach the speed of sound, producing a shock wave with large temperature and pressure gradients in the heat pipe.

This condition can be expressed by the following equation:

$$q_s = 0.474\ L\ \sqrt{\rho v\ Pv} \qquad \text{(Eq. 3)}$$

L = latent heat of vaporization

$\rho v$ = vapor density

$P_v$ = vapor pressure

$q_s$ = heat flux at which the sonic limit is encountered

It can be seen from Equation 3 that the sonic limit will vary with temperature as the vapor properties vary with temperature. The minimum sonic limit will be at the minimum operating temperature. This calculation is carried out in Appendix C, with the result presented below.

$$q_{sonic} = 3456\ \frac{W}{cm^2} \qquad \text{(Eq. 4)}$$

The proof of principle heat pipe will be operating with an axial heat flux below 200 W/cm$^2$ at its maximum. This is well below the sonic limit.

2.3.3.3 Vapor Shear Limit

As the vapor moves down the heat pipe from the evaporator to the condenser, a shear force exists between the liquid in the wick and this vapor. If the vapor velocity is sufficiently high, the shear force exerted on the liquid may impede the flow through the wick. The lack of fluid returning to the evaporator will initiate dryout.

$$q_{vapor\ shear} = L\sqrt{\frac{2\pi\rho_v\sigma\cos\theta}{\lambda}} \qquad \text{(Eq. 5)}$$

L = latent heat of vaporization

$\rho_v$ = vapor density

$\sigma$ = liquid surface tension

$\theta$ = wetting angle

$\lambda$ = characteristic length of vapor-liquid interface (pore diameter)

$q_{vapor\ shear}$ = the heat flux at which the effects of vapor shear begin

Again, as in the sonic limit case, the vapor shear limit will be a minimum when the operating temperature of the heat pipe is a minimum. This calculation is carried out in Appendix D for two configurations, a powder metal wick and a screen wick. The results are presented below:

$q_{vapor\ shear} = 8525\ \frac{W}{cm^2}$ Powder Metal Wick

$q_{vapor\ shear} = 4791\ \frac{W}{cm^2}$ Screen Wick

This shows the vapor shear limit is well above the maximum axial heat flux at which the pipe will be operating.

2.3.3.4 Evaporation Limit

In the evaporator section of the heat pipe a $\Delta T$ exists between the wick structure and the vapor. Because of this temperature difference, there also exists a pressure difference. The pressure created in the capillary wick structure must always be greater than the vapor pressure above the wick because it is undesireable to have vapor bubbles inside the wick. if vapor spaces exist in the wick, hot spots may form which will lead to dryout at the wick.

The major difference between the evaporative limit and the other limits is that all the others were concerned with the axial heat flux, the evaporative limit is concerned with the radial heat flux in the evaporator of the heat pipe. The evaporative limit can be expressed as follows:

$$\frac{K_e\, T_v}{L \rho_v \ln\left(\frac{r_i}{r_v}\right)\left(\frac{r_i + r_v}{2}\right)} \left( \frac{2\sigma}{r_n} - P_c \right) \qquad \text{(Eq. 6)}^{3}$$

$K_e$ = wick thermal conductivity

$T_v$ = vapor temperature

$\sigma$ = surface tension

$r_n$ = nucleation radius of vapor bubbles

$r_i$ = internal radius of pipe

$r_v$ = vapor space radius

$\rho_v$ = vapor density

$L$ = latent heat of evaporation

$P_c$ = capillary pressure $P_c = \frac{2\sigma \cos\theta}{r_c}$

The result of this calculation is presented below, with the calculation shown in Appendix E.

$$q_{\text{evaporative}} = 25.8 \ \frac{W}{cm^2}$$

[3] "Heat Pipes," Dunn, P.D., Pergamon Press, New York, 1982.

As predicted, this limit is well below the maximum radial heat flux at which this pipe will operate. The maximum expected radial heat flux is 3 W/cm$^2$.

### 2.3.4 Heat Pipe and Gas Reservoir Sizing

The heat pipe diameter and length were based on the requirements made clear during Task 1. Additional adiabatic lengths were added between heat sinks to allow for insulation and radiation shields. This prevents excess losses from occurring between the individual components on the heat pipe. Figure 9 illustrates the location of the individual components on the proof of principle heat pipe fabricated.

The gas reservoir size is a function of the volume the gas must occupy while operating in each different mode meeting the operating requirements. The size is also a function of the degree of temperature sensitivity required of the heat pipe overtime. The process and calculations used in finding the reservoir volume can be found in Appendix F, and the results are as follows:

Required Reservoir Volume 72.3 cm$^3$ (4.41in$^3$)

## 2.4 Task 3 - Heat Pipe Fabrication and Test

The purpose of Task 3 was to physically demonstrate that a gas controlled heat pipe, operating within the required specifications, could be fabricated and tested. With this evidence, the feasibility in the application of heat pipes to RTG assemblies would be substantiated. What the proof of principle heat pipe actually demonstrated was the ability to build and operate a device which would remain nearly isothermal while the input heat load was substantially varied.

The configuration of the heat pipe that was fabricated is illustrated in Drawing No. A-202-12. The test setup is much simpler than the system design concept discussed earlier in this report. This alteration allowed

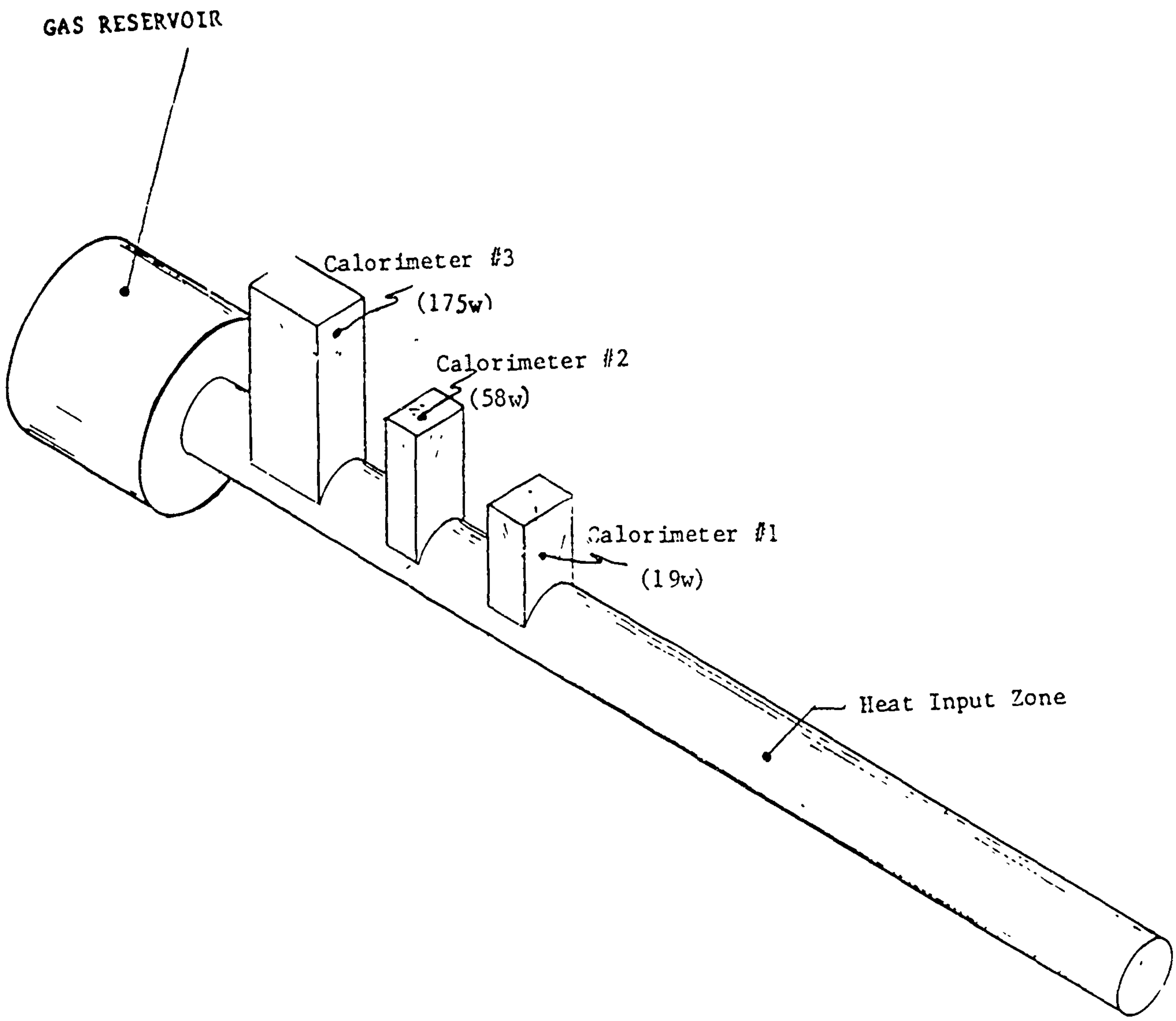



Figure 9. Layout of Proof of Principle Heat Pipe

for more accurate monitoring and recording of the heat pipes operating characteristics.

The heat was be supplied to the evaporator by a cartridge heater placed in the copper cylinder. This method of heating will simulates the heat supplied by an isotopic heat source. The system is similar to the ETG (Electric Thermoelectric Generator) used to model RTG systems. Table 7 provides a list of materials used in the construction of the proof of principle heat pipe.

TABLE 7. MATERIALS OF CONSTRUCTION

| | |
|---|---|
| Envelope and reservoir material | Low carbon steel |
| Wick material | Iron powder metal |
| Working fluid | Mercury |
| Calorimeter material | Low carbon steel |
| Non condensible control gas | Argon |
| Heat source | 500W cartridge heater |

After receipt of all materials, the heat pipe was fabricated, leak checked and charged with mercury. The heat pipe was then outfitted with thermocouples and placed in the test setup shown in Figure 10. Testing was done with the heat pipe in several different orientations from vertical (evaporator down) to horizontal. Samples of the recorded data are presented in the following tables (8, 9, 10, and 11).

Examination of each of the operating conditions clearly illustrates the potential of the gas controlled heat pipe. The heat pipe demonstrated approximately a 6:1 turn down ratio with a 20$^{o}$C heat pipe temperature change. By enlarging and thermally separating the gas reservoir from the heat source, a turn down ratio of 4:1 with a 1$^{o}$C heat pipe temperature change was demonstrated.

TABLE 8. VERTICAL OPERATION OF GAS CONTROLLED HEAT PIPE

| Evaporator | HEAT PIPE TEMPERATURES ($^{o}C$) Adiabatic Zone | | | Reservoir | POWER OUTPUT (W) Calorimeter Number | | | TOTAL POWER Output (W) | TURN DOWN Ratio/$\triangle T$ |
|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | | 1 | 2 | 3 | | |
| 290 | 290 | 269 | 138 | 92 | 28 | 80 | 106 | 214 | -- |
| 288 | 288 | 265 | 103 | 83 | 28 | 78 | 56 | 162 | 1.3:1/2 |
| 278 | 278 | 100 | 31 | 27 | 26 | 47 | 9 | 82 | 2.6:1/12 |
| 271 | 155 | 36 | 21 | 22 | 22 | 11 | 4 | 37 | 5.8:1/19 |

TABLE 9. VERTICAL OPERATION OF GAS CONTROLLED HEAT PIPE WITH ENLARGED, ISOTHERMAL GAS RESERVOIR

| Evaporator | HEAT PIPE TEMPERATURES ($^{o}C$) Adiabatic Zone | | | Reservoir | POWER OUTPUT (W) Calorimeter Number | | | TOTAL POWER Output (W) | TURN DOWN Ratio/ T |
|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | | 1 | 2 | 3 | | |
| 270 | 270 | 251 | 201 | 20 | 26 | 73 | 181 | 280 | -- |
| 269 | 270 | 224 | 161 | 20 | 26 | 72 | 63 | 161 | 4.3:1/1 |
| 267 | 161 | 35 | 24 | 20 | 22 | 11 | 4 | 37 | 7.6:1/3 |

TABLE 10. OPERATION OF GAS CONTROLLED HEAT PIPE AT GRAVITY ANGLE OF $7^{o}$ (EVAPORATOR DOWN)

| | HEAT PIPE TEMPERATURES ($^{o}$C) | | | | POWER OUTPUT (W) | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Evaporator | Adiabatic Zone | | | Reservoir | Calorimeter Number | | | TOTAL POWER Output (W) | TURN DOWN Ratio/$\Delta$T |
| | 1 | 2 | 3 | | 1 | 2 | 3 | | |
| 355 | 348 | 340 | 63 | 29 | 40 | 85 | 39 | 164 | -- |
| 353 | 345 | 175 | 46 | 32 | 39 | 62 | 28 | 129 | 1.3:1/2 |
| 351 | 304 | 101 | 36 | 32 | 39 | 32 | 14 | 85 | 1.9:1/4 |
| 335 | 126 | 44 | 28 | 33 | 23 | 11 | 5 | 39 | 4.2:1/20 |

TABLE 11. HORIZONTAL OPERATION OF GAS CONTROLLED HEAT PIPE

| | HEAT PIPE TEMPERATURES ($^{o}$C) | | | | POWER OUTPUT (W) | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Evaporator | Adiabatic Zone | | | Reservoir | Calorimeter Number | | | TOTAL POWER Output (W) | TURN DOWN Ratio/$\Delta$T |
| | 1 | 2 | 3 | | 1 | 2 | 3 | | |
| 357 | 325 | 83 | 30 | 30 | 40 | 37 | 0 | 77 | -- |
| 347 | 141 | 35 | 25 | 27 | 31 | 11 | 0 | 11 | 1.8:1/10 |
| 124 | 43 | 20 | 22 | 24 | 6 | 4 | 0 | 10 | 7.7:1/233 |

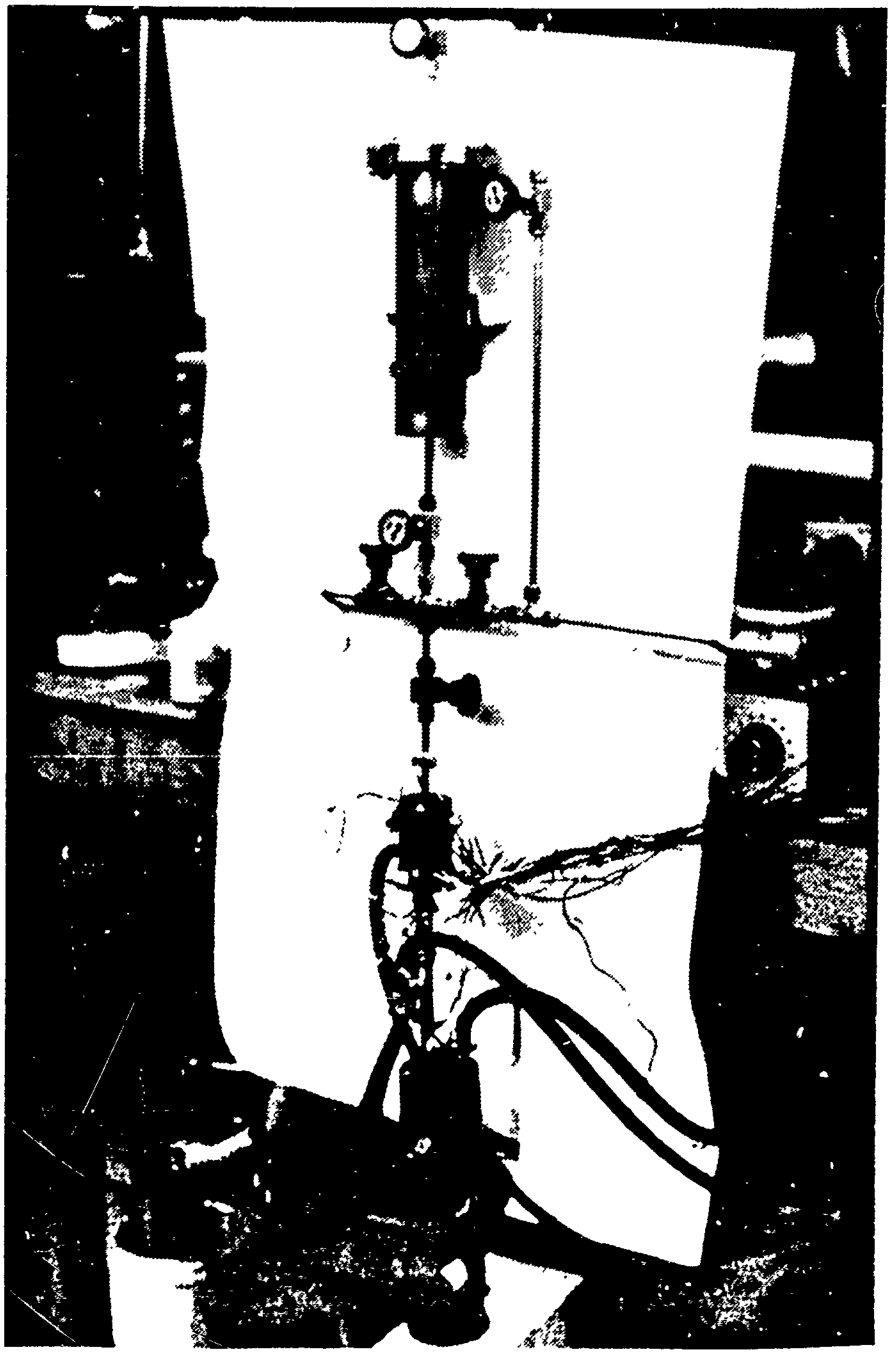


Figure 10

Mercury Heat Pipe Test Setup

Comparison with the predicted operating conditions in Table 7 show that the heat pipe did not behave exactly as was expected. It is believed that the basis for this difference lies in the failure to completely account for the thermal effects on the reservoir in these predictions. This belief was reenforced by the enhanced operation demonstrated by thermal separating the reservoir.

When the heat pipe was operated vertically, it was run to the maximum capabilities of the heaters. In this orientation the heat pipe carried 360 watts. At no time during the test did the heat pipe demonstrate signs of reaching its maximum capabilities. During horizontal operation the heat pipe power handling capabilities were limited. At a power level of approximately 80 watts, the heat pipe began to show signs of dryout. Dryout occurs when the liquid return capabilities of the wick structure are not sufficient to sustain the required rate of evaporation. The wick will literally become void of fluid in the evaporator section of the heat pipe leading to an overheating of the wick structure in that area. Once this occurs, fluid does not wet the area since the fluid is vaporized before it can reach the center of the affected area. In turn, the affected area will continue to expand until the power being supplied to the heat pipe is turned off and the wick allowed to cool. This problem can be eliminated from the present heat pipe design if a wick structure with a higher fluid return capability were used. Such a wick structure would be one with grooves or tunnels. Several reasons existed for not using these more complex wick structures in the present heat pipe. First, horizontal operation of the heat pipe is not one of the necessary provisions of operation. Secondly, it was felt that the simpler the design, the more easily and convincingly the interesting capabilities of the gas controlled heat pipe could be shown.

In addition to the mercury heat pipe, a second heat pipe with Cesium as the working fluid was fabricated and tested. although the heat pipe was not a gas controlled version, it was made in the same configuration as the mercury pipe less the gas reservoir. The purpose of building this pipe was to find the power capabilities of Cesium in this orientation in the event it becomes necessary to use Lead/Telluride thermoelectrics in place of the Bismuth/Telluride. The reason it is necessary to use a different working fluid in each case is the widely different operating temperatures for these two types of thermoelectric modules. The results of the testing on the Cesium heat pipe are presented in Table 12.

Operation of the cesium heat pipe was satisfactory during the testing period of over one hundred hours of operation. However, when the test setup was dismantled, the heat pipe showed signs that a compatibility problem between cesium and iron exists. Even if this is true, it presents no problem to this program because a material switch to stainless steel for the envelope will eliminate any such compatibility problems without sacrificing any operational capabilities.

TABLE 12. CESIUM HEAT PIPE OPERATING CHARACTERISTICS

| HEAT PIPE TEMPERATURES ($^{o}C$) | | | POWER OUTPUT (W) |
|---|---|---|---|
| Evaporator | Adiabatic Zone | Condenser | |
| 590 | 582 | 574 | 46 |
| 595 | 582 | 563 | 46 |
| 589 | 581 | 554 | 45 |

# 3.0 CONSLUSIONS

The successful demonstration of the gas controlled heat pipe designed, fabricated and tested during this program shows that the ability exists to build an RTG assembly powered by a short-lived isotope. This program concluded with the selection of reference components and a potential design for such a system. In summary, these choices were:

| | |
|---|---|
| Isotope Heat Source | Promethium 147 |
| Thermoelectric Module | Bismuth-Telluride |
| Design Model Configuration | Single heat pipe system with a thermoelectric module mounted on a tungsten accumulator block. |

A cost effective design using a short-lived isotope has definite advantages over the currently used design with Plutonium. This becomes evident when the system life of 5 to 10 years is compared to Plutonium's half life of 86 years.

Therefore, the goal of any continued work in the immediate future should be to evaluate the potential for such a system. Before any continued design work is undertaken, the following preliminary tasks should be completed:

1) Establish a meaningful cost estimate for the production of a heat pipe - RTG assembly powered by a low-level isotope.
2) Reevaluate isotopic heat sources. Develope an accurate record of availability, and if necessary, analyze any additional costs which would be incurred for reprossessing and encapsulation facilities.

3) Perform an indepth comparison on the basis of cost, safety and reliability between a Plutonium 238 powered RTG and a RTG powered by a low-level isotope.

Only after this analysis is completed, with favorable results, would the design of any system components be considered.

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3. Sax, N. I. Dangerous Properties of Industrial Materials, Litton Publishing Corp., New York, 1968.

# APPENDIX A

Appendix

THERMACORE, INC. HEAT TRANSFER SPECIALISTS / ENGINEERING—MANUFACTURING
780 EDEN ROAD / LANCASTER, PENNSYLVANIA 17601

Title: Isotope Selection
Project: #202

Calculated by: J. Toth Date: 2/17/84
Checked by: Date:
Reviewed by: Date:



Purpose:

To evaluate the various heat source isotopes on the basis of system Life and the volume each would occupy when used to power a thermoelectric generator supplying 1 $W_e$.

Assumptions:

- System Life: 2 isotope half lives, min. 1 year
- 58 $W_t$ output from isotope capsule required at conclusion of systems' life
- Bismuth-Telluride thermoelectric

## Cobalt-60

### Parameters.

Half · Life = $T_{1/2}$ = 5.24 years

Melting Point: 1495°C

Compound Density = $\rho_{COMP} = 8.8\ g/cm^3$

Compound Power Density: $PD = 15.7\ \frac{W}{cm^3}$

### Rate Constant:

$$k = \frac{.693}{T_{1/2}}$$

$$k = \frac{.693}{5.24} = 0.1323\ \text{years}^{-1}$$

### Final Concentration:

$$A = \frac{\text{Required Final Thermal Output}}{\text{Watts/Gram of compound}}$$

Title: ______ Calculated by: ______ Date: ______
Checked by: ______ Date: ______
Reviewed by: ______ Date: ______
Project: ______



$$A = \frac{58\,W_t}{\left(15.7\,\frac{W}{cm^3}\right)\left(\frac{1}{8.8\,\frac{g}{cm^3}}\right)}$$

$$A = 32.5\,g$$

Initial Concentration:

$$\log \frac{A_o}{A} = \frac{t\,k}{2.3}$$

$$\log \frac{A_o}{32.5g} = \frac{(2)(5.24\,years)(0.1323\,years^{-1})}{2.3}$$

$$A_o = 130.2\,g$$

Capsule Internal Volume.

$$V = \frac{A_o}{\rho_{comp}}$$

$$V = \frac{130.2\,g}{\left(8.8\,\frac{g}{cm^3}\right)} = 14.8\,cm^3$$

Title: ______ Calculated by: ______ Date: ______
Checked by: ______ Date: ______
Reviewed by: ______ Date: ______
Project: ______



| Isotope | $^{60}Co$ | $^{147}Pm$ | $^{170}Tm$ | $^{242}Cm$ |
|---|---|---|---|---|
| System Life (years) | 10.5 | 5.24 | 0.70 | 0.90 |
| Rate Constant (years$^{-1}$) | 0.132 | 0.265 | 1.98 | 1.54 |
| Final Concentration (grams) | 32.5 | 212.5 | 48.3 | 6.64 |
| Initial Concentration (grams) | 130.2 | 853.3 | 193.4 | 26.6 |
| Capsule Internal Volume (cm$^3$) | 14.8 | 129.3 | 24.2 | 2.3 |
| System Life / Capsule Volume (years/cm$^3$) | 0.71 | 0.041 | 0.029 | 0.39 |

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Title: Calculation of Operating Limits
Project: 202

Calculated by: J. Toth  Date: 3/15/84
Checked by:  Date:
Reviewed by:  Date:



Purpose

Appendices B, C, D and E are cumulatively the calculation of the operating limitations on the proof of principle heat pipe being fabricated.

The calculations which are performed are.

Capillary Wicking Height Limit —— Appendix B
Sonic Limit —— Appendix C
Vapor Shear Limit —— Appendix D
Evaporation Limit —— Appendix E

All calculations done with mercury as the selected working fluid.

# APPENDIX B

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| | |
|---|---|
| Title: Capillary Wicking Height Limit | Calculated by: ______ Date: ______ |
| | Checked by: ______ Date: ______ |
| | Reviewed by: ______ Date: ______ |
| Project: ______ | Page 1 of 1 |

## Capillary Wicking Height Limit

- Working fluid must be able to wick height of evaporator to prevent overheating.
- Mercury Working Fluid

$$r_c = \frac{2\sigma \cos\theta}{\rho g h} \qquad (1)$$

$\sigma$ - surface tension - 465 dyne/cm

$\theta$ - wetting angle - 57°

$\rho$ - liquid density - 13.546 g/cm³

$g$ - acceleration of gravity - 981 cm/s²

$h$ - required wicking height - 25.4 cm

$$r_c = \frac{2(465\ \mathrm{dyne/cm})(\cos 57^\circ)}{(981\ \mathrm{cm/s^2})(13.546\ \mathrm{g/cm^3})(25.4\ \mathrm{cm})}$$

$r_c = 0.0015$ cm

APPENDIX C

Appendix C



Title: Sonic Vapor Limit

Project: 062

Calculated by: J Toth  Date: 3/15/84
Checked by:  Date:
Reviewed by:  Date:

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## Sonic Limit

$$q_s = 0.474\, L \left(\rho_v P_v\right)^{1/2}$$

- values at minimum operating temperature ~ 290°C
- mercury working fluid

$L$ - latent heat of vaporization - 72.2 cal/g

$\rho_v$ - vapor density - $1.5 \times 10^{-3}$ g/cm³

$P_v$ - vapor pressure - $3.88 \times 10^{5}$ dyne/cm²

$$q_{sonic} = 0.474\left(72.2\ \tfrac{cal}{g}\right)\left[\left(1.5\times10^{-3}\ \tfrac{g}{cm^3}\right)\left(3.88\times10^{5}\ \tfrac{dyne}{cm^2}\right)\left(\frac{1\ \frac{g\cdot cm}{s^2}}{dyne}\right)\right]^{1/2}$$

$$q_{sonic} = 825.6\ \frac{cal/s}{cm^2} = 3456\ \frac{W}{cm^2}$$

APPENDIX D

Appendix D

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| | | | |
|---|---|---|---|
| Title: Vapor Shear Limit | Calculated by: J [illegible] | Date: 3/15/84 | |
| | Checked by: | Date: | |
| | Reviewed by: | Date: | |
| Project: 202 | | Page 1 of 1 | |

## Vapor Shear Limit

$$q_{\substack{vapor\\shear}} = L \sqrt{\frac{2\pi \rho_v \sigma_\ell \cos\theta}{\lambda}}$$

$L$ - latent heat of vaporization - 72.2 cal/g

$\rho_v$ = vapor density - $1.5\times10^{-3}$ g/cm³

$\sigma_\ell$ = liquid surface tension - 465 dyn/cm

$\theta$ = wetting angle - 57°

$\lambda$ = characteristic length $\begin{cases} \text{powder metal wick - .003 cm} \\ \text{screen mesh wick - .0095 cm} \end{cases}$

$$q_{\substack{vapor\\shear\\(powder\ metal)}} = 72.2\,\frac{cal}{g}\sqrt{\frac{2\pi(1.5\times10^{-3}\ g/cm^3)(465\ dyn/cm)(\cos 57°)}{.003\ cm}}$$

$$q_{\substack{vapor\\shear\\(powder\ metal)}} = 2036.5\ \frac{cal/s}{cm^2} = 8525\ \frac{W}{cm^2}$$

$$q_{\substack{vapor\\shear\\(screen)}} = 4791\ \frac{W}{cm^2}$$

APPENDIX E

Appendix E

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Title: Evaporation Limit

Project: ΔT

Calculated by: J Toth Date: 3/15/84

Checked by: Date:

Reviewed by: Date:

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## Evaporation Limit

$$q_{evap} = \frac{K_e T_v}{L \rho_v \ln\left(\frac{r_i}{r_v}\right)\left(\frac{r_i + r_v}{2}\right)} \left(\frac{2\sigma}{r_n} - P_c\right)$$

$K_e$ - wick thermal conductivity - $.1736 \frac{cal/s}{cm \cdot °K}$

$T_v$ - vapor temperature - 563K

$\sigma$ - liquid surface tension - 465 dyne/cm

$\rho_v$ - vapor density - $1.5 \times 10^{-3}$ g/cm³

$L$ - latent heat of vaporization - 72.2 cal/g

$P_c$ - capillary pumping pressure $\left\{ P_c = \frac{2\sigma \cos\theta}{r_c} = 3.37 \times 10^5 \frac{dyne}{cm^2} \right.$

$r_n$ - nucleation radius of vapor bubbles - $2.54 \times 10^{-3}$ cm (constant)

$r_i$ - internal radius of pipe - .5461 cm

$r_v$ - vapor space radius - .4495 cm (assume 0.03" thick wick)

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Title: ______ Calculated by: ______ Date: ______

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Reviewed by: ______ Date: ______

Project: ______



$$q_{evap} = \frac{\left(.1736 \frac{cal/s}{cm\,K}\right)(563k)\left(1\times10^{-7} \frac{m^2}{cm^2}\frac{kg}{g}\right)}{\left(72.2 \frac{cal}{g}\right)\left(1.5\times10^{-3} \frac{g}{cm^3}\right) \ln\left(\frac{.5461}{.4445}\right)\left(\frac{.5461 + .4445}{2}\right)} \left(\frac{2\left(465 \frac{dyne}{cm}\right)}{2.54\times10^{-3}\,cm} - 3.37\times10^{5} \frac{dyne}{cm^2}\right)$$

$$q_{evap} = 25.8 \frac{W}{cm^2}$$

APPENDIX F

Title: Gas Reservoir Sizing
Project: 202

Calculated by: J Toth Date: 3/1/84
Checked by: Date:
Reviewed by: Date:



Purpose

To formulate an equation from which the required reservoir volume can be calculated for a gas controlled heat pipe operating within the required specifications.

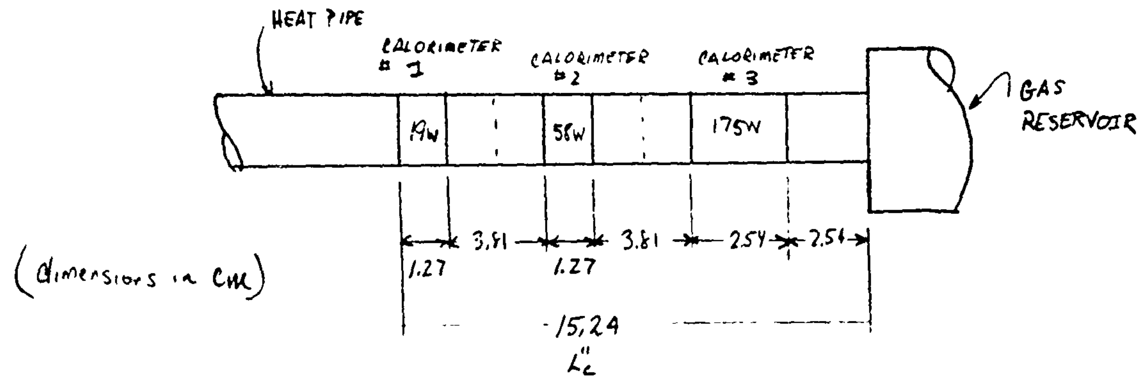
Heat Pipe Operating Modes

| Temperature | Vapor Pressure | Active Length of Condenser |
|---|---|---|
| 290 °C | $3.88 \times 10^5$ $\frac{dyne}{cm^2}$ | 15.24 cm |
| 295 °C | $4.14 \times 10^5$ $\frac{dyne}{cm^2}$ | 8.26 cm |
| 302 °C | $4.62 \times 10^5$ $\frac{dyne}{cm^2}$ | 3.16 cm |

Title: ____________
Project: ____________

Calculated by: ____________ Date: ______
Checked by: ____________ Date: ______
Reviewed by: ____________ Date: ______






In general, heat transfer from a surface can be expressed as:

$$Q = h(A'L)(T_1 - T_2) \qquad (1)$$

$Q$ - rate of heat transfer

$h$ - heat transfer coefficient

$(A'L)$ - heat transfer surface area

$T_1$ - heat pipe temperature

$T_2$ - ambient temperature.

Two equations can be written from the general equation. One at time zero with all gas in the reservoir and one at the end of life when the interface is at its desired position.

at $t=0$

$$Q = Q_{max} = h(A' L_c)(T_{MAX} - T_a) \qquad (2)$$

at $t = t_{end}$

$$Q = Q_{min} = h(A' L_A)(T_{min} - T_a) \qquad (3)$$

$L_A$ - active condenser length

$(L_C - L_A)$ - movement of gas interface

Now, for a perfect gas, the number of moles of gas in the heat pipe and reservoir are always constant

$$dn = \frac{P_g}{R_u T_g} dV \qquad (4)$$

for constant gas pressure and gas temperature

# of moles $\propto$ volume

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Project: ______________________



$$n = \frac{P_g}{R_u T_g}\left[(L_c - L_A)A_v + V_R\right] \qquad (5)$$

$P_g$ - gas pressure in heat pipe ; $P_g = P_{ACTIVE} + P_{...}$

$R_u$ - universal gas constant

$T_g$ - gas temperature

$L_c$ - condenser length

$L_A$ - active condenser length

$A_v$ - area of vapor space (cross-sectional)

$V_R$ - reservoir volume

Solving eq (5) for the active condenser length

$$L_A = L_c - \left\{ \frac{n R_u T_g}{P_g} - V_R \right\} \frac{1}{A_v} \qquad (6)$$

using this active length two additional equations, similar to eq (2) and (3) can be written:

$$Q_{max(2)} = h\left[A'\left(L_c + \frac{V_R}{A_V} - \frac{nR_u T_a}{P_{max}}\right)\right](T_{max} - T_a) \qquad (7)$$

$$Q_{min(2)} = h\left[A'\left(L_c + \frac{V_R}{A_V} - \frac{nR_u T_a}{P_{min}}\right)\right](T_{min} - T_a) \qquad (8)$$

equating eq (2) to eq (7) and solving for n

$$n = \frac{V_R}{A_V}\left[\frac{P_{max}}{R_u T_a}\right] \qquad (9)$$

Similarly with eq (3) and eq (8)

$$n = \left[(L_c - L_A) + \frac{V_R}{A_V}\right]\frac{P_{min}}{R_u T_a} \qquad (10)$$

Because the [illegible] of gas in a closed volume are constant, eq(5) can be set [illegible] eq(1). Doing [illegible] solving for $V_R$ yields:

$$V_R = \frac{(L_C - L_A)A_v}{\left[\frac{P_{max}}{P_{min}} - 1\right]}$$

with;

$L_C$ - condenser length

$L_A$ - active condenser length

$A_v$ - cross-sectional area of vapor space

$P_{MAX}$ - maximum vapor pressure of working fluid at $T_{...}$

$P_{MIN}$ - minimum vapor pressure of working fluid at $T_{...}$

Using the dimensions shown in figure 1, in addition to the operating parameters listed below, the following calculations for reservoir volume and gas interface movement were made

$L_C = 15.24\ cm$

$L_A = 8.26\ cm$

$A_V = 0.694\ cm^2$

$P_{MAX} = 4.14 \times 10^5\ dyne/cm^2$

$P_{MIN} = 3.88 \times 10^5\ dyne/cm^2$

$$V_R = \frac{(L_C - L_A)\,A_V}{\left(\frac{P_{MAX}}{P_{MIN}} - 1\right)}$$

$$= \frac{(15.24 - 8.26)cm\,(.694\,cm^2)}{\left(\frac{4.14 \times 10^5 \frac{dyne}{cm^2}}{3.88 \times 10^5 \frac{dyne}{cm^2}} - 1\right)}$$

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$V_R = 72.29\,cm^3$

Using this reservoir volume and the location of the gas interface needed to demonstrate the second phase of opration, the required vapor pressure was calculated.

$$P_{max} = \left[ \frac{(L_C - L_A)A_V}{V_R} - 1 \right] P_{min}$$

$$= \left[ \frac{(15.24 - 3.16)cm(.694\,cm^2)}{72.29\,cm^3} + 1 \right] 4.14 \times 10^5 \frac{dyne}{cm^2}$$

$$= 4.62 \times 10^5 \frac{dyne}{cm^2}$$

Title: ______ Calculated by: ______ Date: ______
Checked by: ______ Date: ______
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Project: ______



This pressure of $4.62\times10^5\ \frac{dyne}{cm^2}$ corresponds to a vapor at 302°C. With this the operating conditions for the heat pipe are known for each phase of operation. They are:

| Temperature of Heat Pipe (°C) | Vapor Pressure of Gas ($\frac{dyne}{cm^2}$) | Active Length of Condenser (cm) |
|---|---|---|
| 290 | $3.88\times10^5$ | 15.24 |
| 295 | $4.14\times10^5$ | 8.26 |
| 302 | $4.62\times10^5$ | 3.16 |